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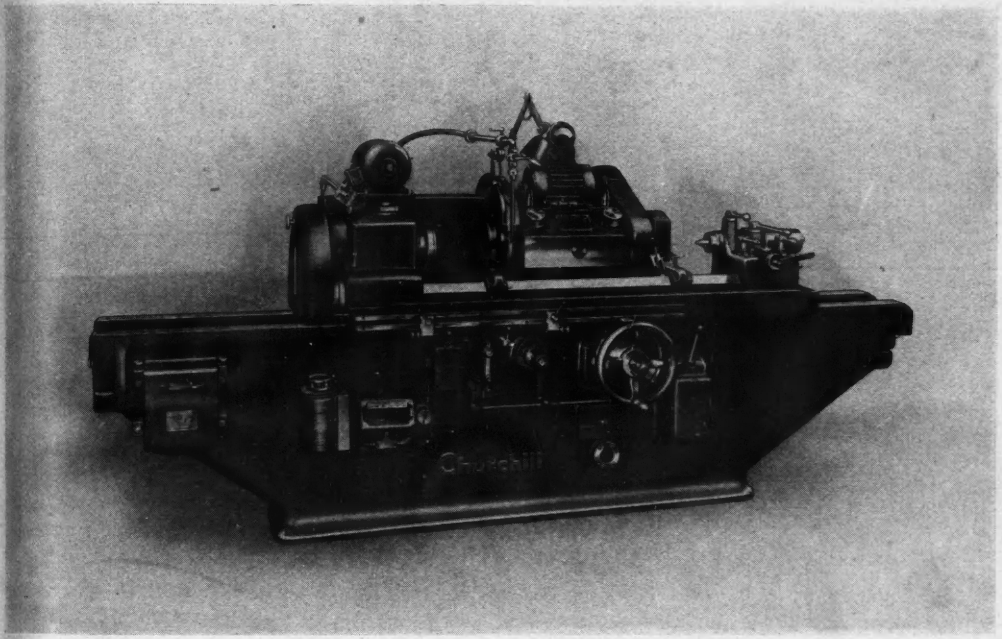

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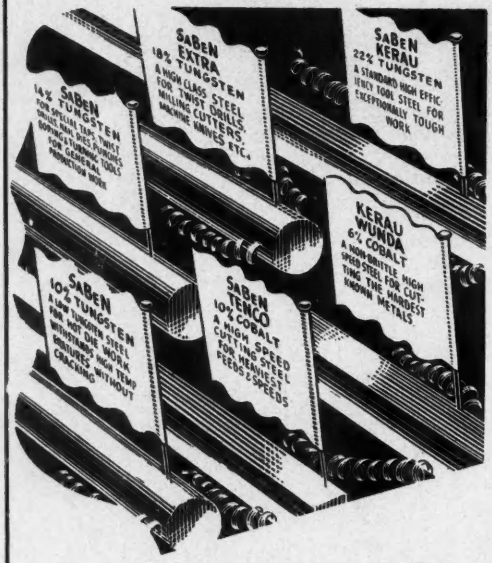

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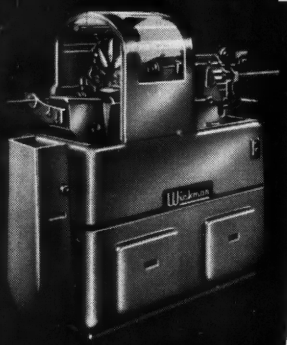

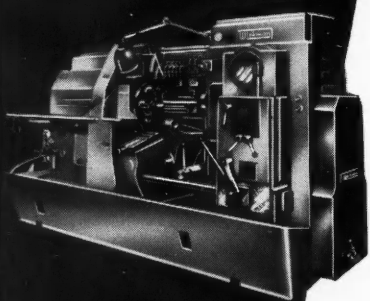

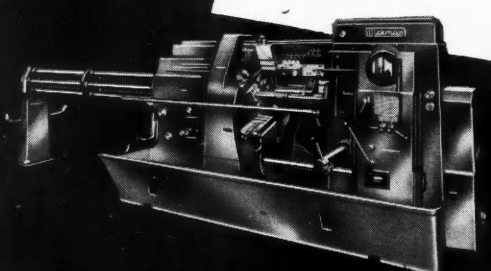

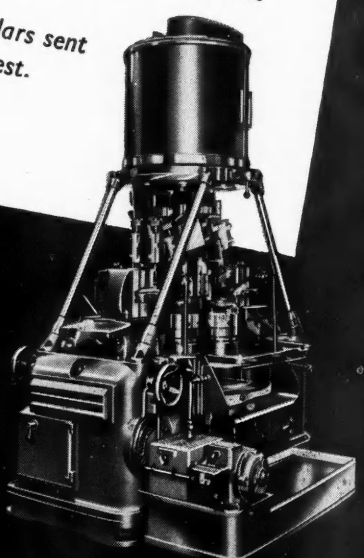

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SYNTHETIC MATERIALS FOR AIRCRAFT CONSTRUCTION.

By E. STEINER. (From *Flugwehr und -Technik*, Vol. 7, No. 11, November, 1945, pp. 281-284, 4 illustrations).

PROGRESS in Aerodynamics together with the development of high-efficiency power plants has raised a demand for greater strength in airframes. It became more and more difficult for the designer to meet the increasing strength requirements with the standard materials at his disposal and it is, therefore, not surprising that he should follow, with interest, the possibilities offered by new structural materials with respect to aerodynamic smoothness of the surface and low structural weight.

PRINCIPLES FOR THE SELECTION OF AIRCRAFT MATERIALS.

Low weight and great strength are the most desirable properties of any material used for aircraft structures. It is customary to compare the different materials with the aid of the "strength-weight coefficient $\frac{\sigma_z}{\gamma}$," where σ_z is the ultimate tensile stress and γ the specific weight. Since members under purely tensile load are rarely encountered in aircraft structures, this coefficient does not lend itself to a general comparison. Therefore, the various coefficients for the most frequent types of loading have been computed and are listed below. The load carrying capacity refers to structural members of different material, but of equal weight. The over-all dimensions of the structural members are also assumed to be equal with the exception of wall thickness which must vary so that equal weights are obtained.

(a) Tensile loading.

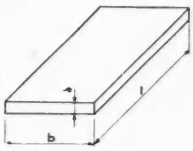
Stresses :

Load carrying capacity $P = \sigma_z \,.\, b \,.\, \delta$

Weight $G = b \,.\, \delta \,.\, l \,.\, \gamma$

$$\delta = \frac{G}{b \,.\, l} \,.\, \frac{1}{\gamma} = k \,\frac{1}{\gamma}$$

$$P = k \,.\, b \,.\, \sigma_z \,.\, \frac{1}{\gamma} = k' \frac{\sigma_z}{\gamma}$$



Elastic Deformations :

$$\Delta l = \frac{\sigma_z}{E} \,.\, l = \frac{P \,.\, l}{\delta \,.\, b \,.\, E}; \quad \text{with } \delta = k \frac{1}{\gamma}$$

it becomes

$$\Delta l = \frac{P \,.\, l^2}{G} \,.\, \frac{\gamma}{E}; \quad P = \frac{\Delta l \,.\, G}{l^2} \,.\, \frac{E}{\gamma} = k'' \,.\, \frac{E}{\gamma}$$

The characteristic coefficients are :

$$\frac{\sigma_z}{\gamma} \text{ and } \frac{E}{\gamma}$$

(b) Bending.

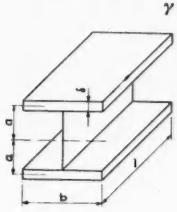
With reference to the I-section below :

Stresses $M_b = \sigma_b \,.\, W = \sigma_b \,.\, 2a \,.\, b \,.\, \delta = k \,.\, \frac{\sigma_b}{\gamma}$

Stiffness (cantilever beam)

Deflection $f = \dfrac{P \,.\, l^3}{3\,\mathfrak{J} E}$

For constant deflection the load carrying capacity is



$$P_b = \frac{3\,\mathfrak{J} E \,.\, f}{l^3};$$

$$\mathfrak{J} = 2\,a^2 \,.\, b \,.\, \delta = k \,.\, \frac{1}{\gamma}; \; P = \frac{3f}{l^3} \,.\, k \,.\, \frac{1}{\gamma} \,.\, E = k' \,.\, \frac{E}{\gamma}$$

Characteristic coefficients for bending :

$$\frac{\sigma_b}{\gamma} \text{ and } \frac{E}{\gamma}$$

(c) Euler buckling

$$P_k = \frac{\pi^2 E \mathfrak{J}}{l^2} = k \,.\, \frac{E}{\gamma}$$

Characteristic coefficient : $\dfrac{E}{\gamma}$

(d) Buckling of shells in compression

(1) Circular cylindrical shells :

From the theory of elacticity the buckling stress is given as

$$\sigma_K = \frac{E}{3\sqrt{1-\nu^2}} \,.\, \frac{\delta}{r} \approx 0{\cdot}6 \,.\, E \,.\, \frac{\delta}{r}$$

where ν = Poissons' ratio.

Due to initial buckling unavoidable in the manufacture the actual buckling stress is lower. But this does not affect the general relationship established by the formula.

Buckling load

$$P_K = k \,.\, E \,.\, \delta \,.\, u \,.\, \delta = k' \,.\, E \,.\, \delta^2 = k'' \,.\, \frac{E}{\gamma^2}$$

(u = peripheral length)

(2) Flat plates

$$\sigma_K = k \,.\, \frac{1}{1-\nu^2} \,.\, E \left(\frac{\delta}{b}\right)^2$$

$$P_K = k' \,.\, E \,.\, \delta^3 = k'' \,.\, \frac{E}{\gamma^3}$$

(e) Buckling of shells due to shear :

(1) Circular cylindrical shells

From the theory of elasticity

$$\tau_K = 0{\cdot}815\, E \left(\frac{r}{l}\right)^{1/2} \left(\frac{\delta}{r}\right)^{5/4}$$

$$\text{for } \frac{l}{r} < 6{\cdot}5 \sqrt{\frac{r}{\delta}}$$

and $\tau_K = 0{\cdot}29\, E \left(\dfrac{\delta}{r}\right)^{3/2}$ for $\dfrac{l}{r} > 6{\cdot}5 \sqrt{\dfrac{r}{\delta}}$

Due to initial buckling the actual values are from 20 to 40 per cent. lower

Buckling load $P_K = k \,.\, E \,.\, \delta^{9/4} = k' \,.\, \dfrac{E}{\gamma^{2{\cdot}25}}$

and $P_K = k \,.\, E \,.\, \delta^{5/2} = k' \,\dfrac{E}{\gamma^{2{\cdot}5}}$

(2) Curved plates

$$\tau_K = 1{\cdot}25 \,\frac{E \,.\, \delta}{b} \left(\frac{\delta}{r}\right)^{1/2}$$

$$P_K = k \cdot E \cdot \delta^{5/2} = k' \cdot \frac{E}{\gamma^{2.5}}$$

(3) Flat plates

$$\tau_K = k \cdot E \left(\frac{\delta}{b}\right)^2$$

$$P_K = k \cdot E \cdot \delta^3 = k' \cdot \frac{E}{\gamma^3}$$

The characteristic coefficients for plate and shell buckling are thus :

$$\frac{E}{\gamma^2}, \quad \frac{E}{\gamma^{2.25}}, \quad \frac{E}{\gamma^{2.5}}, \quad \frac{E}{\gamma^3}$$

CRITICAL ANALYSIS OF SYNTHETIC MATERIALS AND A COMPARISON WITH STANDARD MATERIALS.

1.—WEIGHT AND STRENGTH.—The characteristic coefficients given above have been calculated for various materials and are listed in Table I.

TABLE I.—MATERIALS AND THEIR PROPERTIES.

Item	Material	Ultimate tensile kg. per sq. cm.	Modulus of elasticity kg. per sq. cm.	Spec. weight
1	High grade steel	12000	2,100000	7·85
2	Avional	4000	720000	2·8
3	Electron	2500	450000	1·8
4	Cardboard[1]	1070	110000	1·37
5	Plywood[1] yellow poplar ; veneer joined with phen. resin	2070	222000	1·26
6	Plywood[2] (beech veneers)			
a	7 veneers per cm.	1250	160000	0·71
b	25 veneers per cm.	1676	172000	0·85
c	100 veneers per cm.	1736	215000	1·10
7	Lignofol[2] beech veneers joined under high pressure	1900	250000	1·37

With materials 4–7, bonding is achieved by means of a phenol-formaldehyde resin.

TABLE II.—ABSOLUTE VALUES OF COEFFICIENTS.

Material	$\frac{\sigma_z}{1000\gamma}$	$\frac{E}{1000\gamma}$	$\frac{E}{1000\gamma^2}$	$\frac{E}{1000\gamma^3}$
1	1·53	268	34·2	4·34
2	1·43	257	91·9	32·9
3	1·39	250	139	77·3
4	0·78	80	58·5	42·6
5	1·64	176	139	110
6a Plywood	1·76	225	318	447
6b Plywood	1·97	202	239	281
6c Plywood	1·58	195	178	161
7	1·39	182	133	97

TABLE III.—COEFFICIENTS BASED ON AVIONAL = 100 PER CENT.

Material	$\frac{\sigma_z}{1000\gamma}$ per cent	$\frac{E}{1000\gamma}$ per cent	$\frac{E}{1000\gamma^2}$ per cent	$\frac{E}{1000\gamma^3}$ per cent
1 Steel	107	104	37·2	13·2
2 Avional	100	100	100	100
3 Electron	97	97·3	151	235
4 Cardboard	55·5	31·1	63·6	129
5 Plywood	115	68·5	151	335
6a Plywood	123	87·6	346	1360
6b Plywood	138	78·6	260	855
6c Plywood	110	76	193	490
7 Lignofol	123	70·9	145	295

The comparison in Table III. gives but an approximate picture of the behaviour of the various materials, because of the following considerations :

Synthetic materials do not follow the law of elasticity as closely as metals. Fletcher[3], however, in his report on the buckling of plastic shells, points out that the general tendency is the same as that for metals.

The characteristic coefficients have been determined for structural members of the same over-all dimensions, despite the fact that designs should be made so as to suit the various materials.

Within these limitations it can be stated that :—

(a) with respect to tensile and bending stresses, all materials considered (with the exception of cardboard), are about equal.
(b) with respect to bending stiffness and freedom from Euler-buckling, synthetic materials are inferior.
(c) synthetic materials offer great advantages from the point of view of weight especially when shell-design is employed. This is clearly seen from the third and fourth columns of Table III.

It may be worth while to mention the strength tests carried out by Küch[4]. The design of the plastic shells, however, was similar to the metal shell-type design (Z and Ω stringers), and therefore these tests cannot be considered to be conclusive. Plastic shells should be developed in such a manner as to bring out the merits of the material employed. The damping capacity of synthetic materials is greater than that of metals[5], an advantage not to be underrated in view of aircraft vibrations.

MANUFACTURING METHODS.

Metal shell constructions, to be sufficiently stiff, require a great number of stringers and stiffeners to support the thin skin. In manufacturing these, a great amount of work (riveting) and time is necessary. Plastic shells dispense with this waste of effort ; on the other hand, the pressures and temperatures involved require special attention. To achieve homogeneous material in plastic design, pressures of 300 to 500 kg. per sq. cm. must be employed. Thus, for a shell of one sq. metre surface a 3000 to 5000 ton press is required.

The various methods of manufacture, such as the Duramold method[6] or the Aeromold method[6] have been extensively treated in literature. As for synthetic-resin-bonded plywood, pressures of only 15–25 kg. per sq. cm. are necessary. This, at present, is the only material suitable for shell-design in the full meaning of the term.

The setting temperature for phenol-formaldehyde resins is 120–180 deg. C. ; the setting time decreases hyperbolically with temperature increase (16 min. at 120 deg. C., and 2 min. at 170 deg. C.)[2]

The absence of rivets and the method of production ensures a much greater aerodynamic smoothness of the plastic shell than is obtainable with stressed-skin metal structures. Moreover, the thick skin of plastic shells is less likely to buckle in the range of normal loading, whereas elastic folds in metal-skin structures appear long before the load limit is reached.

CONCLUSIONS.

A comparison of synthetic materials with metals with regard to weight, strength, manufacture, surface condition, etc., appears to indicate that synthetic materials may offer certain advantages, especially synthetic-resin-bonded veneers. Further research into the technique of bonding should be undertaken and the most suitable design for synthetic materials has yet to be established.

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U. S. A.

THE STRENGTH OF PLYWOOD.

An Outline of the Investigations at the Forest Products Laboratory, Madison 5, Wisconsin, During World War II

By CHARLES B. NORRIS, Forest Products Laboratory, Madison, Wis., U.S.A.

WHEN war was declared in 1941 a group of research men headed by Dr. H. W. March were investigating the strength and elastic stability of plywood panels at the U.S. Forest Products Laboratory at Madison 5, Wisconsin. The writer joined this group in February, 1942. Shortly thereafter the Laboratory was asked to prepare a handbook covering the strength of wood and plywood for use by designers of wood aircraft. A summary of the information available was published the following July in the "ANC Handbook on the Design of Wood Aircraft Structures."

The preparation of this publication was under the auspices of the Army, the Navy, and the Civil Aeronautics Administration through the Army-Navy-Civil Committee on Aircraft Design Criteria. Its preparation focused attention on numerous critical inadequacies in the available information and an expanded programme of investigative work was undertaken. The handbook was revised and republished in June 1944 as ANC Bulletin 18, "Design of Wood Aircraft Structures," to include data available to that date. The work is still going forward and further revisions and additions to ANC 18 are contemplated.

The ANC Bulletin gives the results of the work on plywood in the briefest form. It is supplemented, however, by detailed reports published by the Forest Products Laboratory. The purpose of this article is to outline the information in these reports and to indicate the scope of the work still under way. Numbers occurring in parentheses throughout this text designate reports published by the Laboratory. These are listed at the end of the article.

The number of different constuctions of plywood which can be manufactured by use of different species and thicknesses of veneers is tremendous. It was necessary, therefore, to relate the strength and elastic properties of plywood to the properties of the wood and the thicknesses of the veneers of which the plywood is made. The investigations undertaken can be grouped under four general headings :

1. Strength and elastic properties of plywood as a material.
2. Flat plates of plywood.
3. Cylinders and curved plates of plywood.
4. Stiffened flat plates, cylinders, and curved plates of plywood.

The information now available is outlined under the first three headings. The work covered by the last heading has not yet been reported.

The plywood considered is that of the normal constructions in which the grain direction of each ply is perpendicular to the grain directions of the adjacent plies and the construction is symmetrical about the central plane of the plywood. In some instances non-symmetrical plywood is considered.

STRENGTH AND ELASTIC PROPERTIES OF PLYWOOD AS A MATERIAL.

It is convenient to consider plywood as a material rather than as a construction of various materials, and to determine its overall strength and elastic properties. The obvious disadvantage of this approach is that the elastic properties are different for stresses induced by flexure than for those directly applied. This disadvantage, however, seems to be overbalanced by the simplicity of the formulas obtained, particularly those having to do with elastic stability.

When stresses are appied lto plywood in directions parallel or perpendicular to the direction of the grain of the faces, the strength and elastic properties of the plywood can be expressed in terms of those properties of the veneers from which it is made (1312)*. Such correlations have been made for compression (1315) flexure (1304), and shear (report not yet published). It was found that the properties are not materially influenced by the presence of glue lines (except in the case of shear) if the individual plies are not too thin. Resin glue impregnates the wood adjacent to the glue lines and raises its stiffness and strength. This effect can be great if the plies are very thin. The effect varies, of course, with the kind of glue used.

The mathematical theory of elasticity suggests methods by which the elastic properties of plywood at angles to the face grain can be determined in terms of the properties parallel and perpendicular to the face grain. These methods have been developed mathematically (1503) and by means of Mohrs' stress and strain circles (1317). The formulæ obtained have been verified by experiment (1328 and supplements A, B, and C). An interaction formula also has been suggested and verified for the determination of the strength in compression, tension, or shear at any angle to the face grain (1328). A method for the determination of the modulus of rigidity under flexure has also been developed (1301).

The general approach to the problem tacitly assumes that the values of the properties of the veneers can be corrected for moisture content and that these corrected values can be used in the calculation of the overall properties of the plywood. This assumption was verified by test (1306, 1519).

A beginning has been made in the study of the fatigue behaviour of plywood. It was found that it is not greatly different from that of wood nor, for that matter, from that of many other materials (1305 ; 1327, and supplement A).

FLAT PLATES OF PLYWOOD.

When plywood is used in a structure it is often in the form of flat plates supported at their edges. The stiffness and strength of such plates when subjected to loads normal to their surfaces are important. The mathematical theory of elasticity leads to useful formulæ (1300, 1312) which were verified by test.

Plywood plates may buckle when subjected to edgewise stresses. The critical stress may be considerably below the strength of the plywood. Formulæ based upon the mathematical theory of elasticity (1507) were derived for a number of important cases (1316 and supplements A, B, C, and F). Those cases for which testing techniques were devised, were verified by experiment (1316 and supplements D, G, and H). An "effective width" method for the determination of the strength of such panels after they had buckled was devised and empirical curves for design purposes were obtained (1316 and supplements E and I).

The buckling of plywood webs in box beams is an important special case. The plywood is subjected to bending in its plane as well as to shear ; also it is not rigidly supported by the flanges of the beam. Special methods of study and design were developed (1318 and supplements A to E, inclusive).

* Figures in brackets refer to the list of reports at the end of this article.

CYLINDERS AND CURVED PLATES OF PLYWOOD.

The buckling of plywood cylinders when subjected to various loading conditions is important because of the analogous use of plywood in aircraft. Axial compression, torsion, bending, and combined bending and torsion have been studied.

A mathematical analysis of plywood cylinders in axial compression was made (1322-A), verified by test (1322-B), and a convenient summary of the work prepared (1322). An experimental determination of the effect of the length of the cylinders was made (1514) and the effect of double curvature (1323). The mathematical results can be readily applied to isotropic materials, such as metals. The resulting formula differs from those previously devised for metals and is in better agreement with experiment.

An analysis of plywood cylinders in torsion was made and verified by test (1529). The mathematical results, again, can be applied to isotropic materials and are in as good, if not better, agreement with experiments as are the formulæ previously devised for metals.

A formula was obtained for cylinders in bending by comparison with the results obtained from the study of compression (1502) and in combined bending and torsion by use of the interaction principle (1501).

Curved plywood plates were considered as partial cylinders and the effect of the support supplied at their edges was determined by test (1508). Curved plywood plates subjected to shear stresses at their boundaries were similarly considered as partial cylinders subjected to torsion (report not yet published).

STIFFENED FLAT PLATES, CYLINDERS AND CURVED PLATES OF PLYWOOD

Plywood may be used in semimonocoque structures. In such structures the thickness of the plywood is chosen so that its stiffness is not sufficient to avoid buckling at high loads. It is necessary, therefore, to stiffen the plywood by gluing strips of wood (stiffeners) to one of its surfaces. Such stiffeners cause each section of the plywood to buckle independently of the adjacent sections and thus increase the stress at which buckling occurs. The stiffeners should be sufficiently heavy to accomplish this purpose, but excess weight serves no purpose except to increase the weight of the structure. The structure should also be elastically stable as a whole.

The Forests Products Laboratory has made considerable progress toward the solution of this problem although the results of the work have not yet been published. A mathematical analysis of the stiffness added to a panel by a stiffener has been made and verified by test. Analyses of the critical stresses of plywood panels subjected to edgewise compressive stresses and stiffened with a single stiffener placed parallel to or perpendicular to the direction of the stress have been made and verified by test. Empirical studies of flat panels and cylinders stiffened with a multiplicity of stiffeners are under way. Edgewise compression tests upon curved panels stiffened by a single stiffener are being made. A study of stiffened cylinders subjected to combined normal pressure and axial compression has been undertaken.

The results of this work are not only applicable to plywood, but to other materials. The mathematical attack makes it possible to put the results in terms of most materials whose properties are known, particularly isotropic materials, such as the metals.

During the war the publications of the results of the Forest Products Laboratory research on plywood were classified as "restricted." These restrictions have been removed and the reports are now available for public distribution.

LIST OF REPORTS.

Report No.	*Title.*
1300	Summary of formulae for flat plates of plywood under uniform or concentrated loads.
1301	Method of measuring the shearing moduli in wood.
1304	Methods of computing the strength and stiffness of plywood strips in bending.
1305	Effect of 5000 cycles of repeated bending stresses on 5-ply Sitka spruce plywood.
1306	Effect of moisure on the compressive strength of plywood and laminated wood.
1312	Flat plates of plywood under uniform or concentrated loads.
1315	Tentative method of calculating the strength and modulus of elasticity of plywood in compression.
1316	Buckling of flat plywood plates in compression, shear, or combined compression and shear.
	Supplements to 1316:
1316-A	Buckling of flat isotropic plates in compression, shear, or combined compression and shear.
1316-B	Buckling of plates of any symmetrical construction. Edges simply supported. Buckling of plates with two edges clamped.
1316-C	Plates having the grain of the face plies inclined to the edges.
1316-D	Buckling of flat plywood plates in compression with face grain at 0 deg. and 90 deg. to load.
1316-E	Effective width of thin plywood plates in compression with the face grain at 0 deg. and 90 deg. to load.
1316-F	Buckling of long flat plywood plates under uniform shear. Grain of face plies inclined to edges. Edges clamped.
1316-G	Buckling tests of flat plywood plates in compression with face grain at 15 deg., 30 deg., 45 deg., 60 deg and 75 deg. to load.
1316-H	Buckling of flat plywood plates in uniform shear with face grain at 0 deg., 45 deg., and 90 deg.
1316-I	Effective width of thin plywood plates at maximum load in compression with the face grain at 0 deg., 30 deg., 45 deg., 60 deg., 75 deg. and 90 deg. to load.
1317	The application of Mohr's stress and strain circles to wood and plywood.
1318	Design of plywood webs in box beams.
	Supplements to 1318:
1318-A	Stiffeners in box beams and details of design.
1318-B	Buckling in shear webs of box and I-beams and the effect upon design criteria.
1318-C	Additional tests of box beams and I-beams to substantiate further the design curves for plywood webs in box beams—tests of plywood webs in the tension field.
1318-D	Buckling and ultimate strengths of shear webs of box beams having plywood face grain direction parallel or perpendicular to the axis of the beams.
1318-E	The effect of repeated buckling on the ultimate strengths of box beams with shear webs in the inelastic buckle range.
1322	Buckling of long, thin, plywood cylinders in axial compression.
	Supplements to 1322:
1322-A	Mathemetical treatment.
1322-B	Experimental treatment.
1323	A comparison of the buckling strength of thin-walled cylindrical and barrel-shaped plywood shells.
1327	The fatigue behaviour of wood and plywood subjected to repeated and reversed bending stresses.
	Supplement to 1327:
1327-A	The fatigue behaviour of Douglas-fir and Sitka spruce subjected to reversed stresses superimposed on steady stresses.
1328	Compression, tension and shear tests on yellow-poplar plywood panels of sizes that do not buckle with tests made at various angles to the face grain.
1328-A	Compression tests.
1328-B	Tension tests.
1328-C	Shear tests.
1501	Thin-walled plywood cylinders in bending and torsion.
1502	Thin-walled plywood cylinders in bending.
1503	Stress-strain relations in wood and plywood considered as orthotropic materials.
1507	Rectangular plywood plates with the grain of the face plies inclined to the edges.
1508	Buckling of thin, curved, plywood plates in axial compression.
1514	Effect of length on the buckling stresses of thin-walled, plywood cylinders in axial compression.
1519	Effect of moisture on the compressive, bending, and shear strengths, and on the toughness of plywood.
1529	Buckling of thin-walled plywood cylinders in torsion.

TORSIONAL RESISTANCE OF AN ANNULAR SECTOR.

By M. HAMPL. (From *Skoda Mitteilungen*, No. 4, 1943, 8 pp., 2 illustrations).

NUMERICAL values of the moment of inertia of an annular sector are not yet known in the technical literature, although the necessary theoretical principles have been partly quoted, e.g., in Love, *Mathematical Theory of Elasticity*, Cambridge, 1927, p. 319 ; and Weber, *Die Lehre der Drehungsfestigkeit* (Forschungsarbeiten a.d.g.d. Ingenieurwesens, Berlin, 1921, p. 41). These numerical values have been computed in this paper by means of digamma-and trigamma-functions.

If M is the twisting couple, G the modulus of rigidity, and ϑ the angle of twist per unit length, then $M = G\,\vartheta\,\mathcal{J}$ where $\mathcal{J}$ is the moment of inertia of the annular sector. $\mathcal{J}$ is a function of the cross-section, i.e., in the case of an annular sector it depends on the values of a, b, and 2α. In Fig. 1 a is the outer radius, b the inner radius, and 2α the central angle of the annular sector. By means of the literature mentioned above we have derived the term $\mathcal{J}$ as follows :

$$\frac{1}{a^4}\mathcal{J} = \frac{1}{4}(1-u^4)(2\alpha - \operatorname{tg} 2\alpha) + \frac{8}{\alpha}\sum_{n=0}^{\infty}\frac{1}{k(k^2-4)}\frac{1}{1-u^{2k}}\left\{\frac{(1-u^{k+2})^2}{k+2} - \frac{(u^2-u^k)^2}{k-2}\right\} \quad .. \quad .. \quad .. \quad .. \quad (1)$$

where

$$u = \frac{b}{a},\ k = \frac{\pi}{2\alpha}(2n+1),\ n = 0, 1, 2 \quad .. \quad (2)$$

It is advantageous for the numerical computation of $\mathcal{J}$ to express the term tg 2α in equation (1) by the following infinite series

$$\operatorname{tg} 2\alpha = \frac{4}{\alpha}\sum_{n=0}^{\infty}\frac{1}{k^2-4} \quad .. \quad .. \quad (3)$$

(See e.g., B. Adams, *Smithonian Mathematical Formulae*, Washington, 1922, p. 129).

$$\frac{1}{a^4}\mathcal{J} = \frac{\alpha}{2}(1-u^4) + \frac{1}{\alpha}\sum_{n=0}^{\infty}A_n \quad .. \quad (4)$$

where (after a transformation)

$$A_n = -\frac{k+4}{k(k+2)^2}(1-u^4) - 32u^4\frac{(1-u^{k-2})^2}{k(k^2-4)^2(1-u^{2k})} \quad .. \quad (5)$$

By denoting

$$S_1(\alpha) = \sum_{n=0}^{\infty}\frac{k+4}{k(k+2)^2} \quad .. \quad .. \quad (6_1)$$

$$S_2(\alpha, u) = \sum_{n=0}^{\infty}\frac{(1-u^{k-2})^2}{k(k^2-4)^2(1-u^{2k})} \quad .. \quad (6_2)$$

equation (4) transforms to :

$$\frac{1}{a^4}\mathcal{J} = \frac{\alpha}{2}(1-u^4) - \frac{1}{\alpha}(1-u^4)S_1(\alpha) - \frac{32u^4}{\alpha}S_2(\alpha, u). \quad .. \quad .. \quad (7)$$

THE FUNCTIONS OF DIGAMMA AND TRIGAMMA.

Before summing up the infinite series S_1 and S_2, we quote some formulae known from the theory of the Gamma-functions.

We denote

$$\psi(x) = \frac{d}{dx}[\log\Gamma(x+1)] \text{ Digamma-function}$$

$$\psi'(x) = \frac{d^2}{dx^2}[\log\Gamma(x+1)] \text{ Trigamma-function}$$

Then

$$\sum_{n=1}^{\infty}\frac{1}{n(n+x)} = \frac{1}{x}[C + \psi(x)], \quad .. \quad (8_1)$$

$$\sum_{n=1}^{\infty}\frac{1}{(n+x)^2} = \psi'(x), \quad .. \quad .. \quad (8_2)$$

where $C = 0{\cdot}5772156649\ldots$ (Euler's Constant).

The tables of the functions $\varphi(x)$ and $\varphi'(x)$ are given in E. Pairman, *Tables of the Digamma and Trigamma-Functions* (Pearson, *Tracts for Computors*, No. I., Cambridge, 1919).

The following formulae for these functions hold good :

$$\left.\begin{aligned}\psi(0) &= -C,\ \psi'(0) = \sum_{n=1}^{\infty}\frac{1}{n^2} = \frac{\pi^2}{6},\\ \psi(x) &= \frac{1}{x} + \psi(x-1),\\ \psi'(x) &= -\frac{1}{x^2} + \psi'(x-1),\end{aligned}\right\} \quad (9)$$

$$\left.\begin{aligned}\psi(nx) &= \log_n n + \frac{1}{n}\left\{\psi\left(x - \frac{n-1}{n}\right) + \psi\left(x - \frac{n-2}{n}\right) + \ldots + \psi\left(x - \frac{1}{n}\right) + \psi(x)\right\},\\ \psi'(nx) &= \frac{1}{n^2}\left\{\psi'\left(x - \frac{n-1}{n}\right) + \psi'\left(x - \frac{n-2}{n}\right) + \ldots + \psi'\left(x - \frac{1}{n}\right) + \psi'(x)\right\}.\end{aligned}\right\} \quad (10)$$

From equation (8_1) a new and necessary relation is obtained :

$$\sum_{n=1}^{\infty}\frac{1}{(n+x)(n+y)} = \frac{1}{x-y}[\psi(x) - \psi(y)] \quad (11)$$

$$\sum_{n=1}^{\infty}\frac{1}{(2n+1)(2n+1+x)} = \frac{1}{x}\left[\frac{1}{2}C + \psi(x) - \frac{1}{2}\psi\left(\frac{x}{2}\right)\right] \quad .. \quad .. \quad .. \quad (12)$$

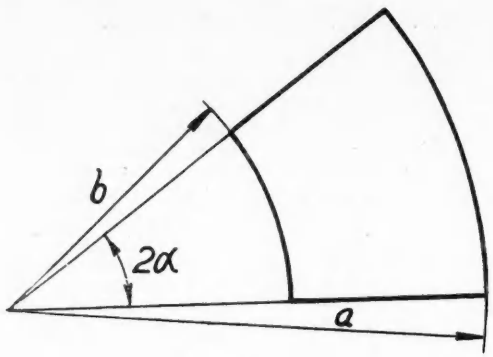


Fig. 1

THE SERIES $S_1(\alpha)$.

The k–th term of the series (6_1) :

$$\frac{k+4}{k(1+2)^2} = \frac{2}{k(k+2)} - \frac{1}{(k+2)^2}$$

reads after the substitution

$$k = \frac{\pi}{2\alpha}(2n+1)$$

$$\frac{k+4}{k(k+2)^2} = \frac{8\alpha^2}{\pi^2}\frac{1}{\left(2n+1\right)\left(2n+1+\frac{4\alpha}{\pi}\right)} - \frac{\alpha^2}{\pi^2}\frac{1}{\left(n+\frac{1}{2}+\frac{2\alpha}{\pi}\right)^2}$$

Summing up both sides of the last equation from $n = 0$ to $n = \infty$, one obtains using equations (12) and (8_3) :

$$S_1(\alpha) = \sum_{n=0}^{\infty}\frac{k+4}{k(k+2)^2} = \frac{2\alpha}{\pi}\left\{\frac{1}{2}C + \psi\left(\frac{4\alpha}{\pi}\right) - \frac{1}{2}\psi\left(\frac{2\alpha}{\pi}\right)\right\} - \frac{\alpha^2}{\pi^2}\left\{\frac{1}{\left(\frac{1}{2}+\frac{2\alpha}{\pi}\right)^2} + \psi'\left(\frac{1}{2}+\frac{2\alpha}{\pi}\right)\right\}. \quad (13)$$

From equations (10) and (9) it follows :

$$\psi\left(\frac{4\alpha}{\pi}\right) = \log 2 + \frac{1}{2}\left[\psi\left(\frac{2\alpha}{\pi}-\frac{1}{2}\right) + \psi\left(\frac{2\alpha}{\pi}\right)\right].$$

If we put

$$C = 2 - 2\log 2 - \psi\left(\frac{1}{2}\right)$$

and denote

$$p = \frac{2\alpha}{\pi}\left[\text{i.e. } k = \frac{1}{p}(2n+1)\right],$$

we get finally

$$S_1(\alpha) = p\left[1 - \frac{1}{2p+1} - \frac{p}{(2p+1)^2}\right] + \frac{1}{2}p\left\{\psi\left(p+\frac{1}{2}\right) - \psi\left(\frac{1}{2}\right) - \frac{1}{2}p\,\psi'\left(p+\frac{1}{2}\right)\right\}. \quad (14)$$

The numerical values of $S_1(\alpha)$ are given in the following table :

TABLE I.

2α	p	$S1(\alpha)$
2π	2	2·1763 0891
π	1	0·7662 9945
$\frac{2}{3}\pi$	$\frac{2}{3}$	0·3972 9272
$\frac{1}{2}\pi$	$\frac{1}{2}$	0·2437 6521
$\frac{1}{3}\pi$	$\frac{1}{3}$	0·1187 9923
$\frac{1}{4}\pi$	$\frac{1}{4}$	0·0699 8927
$\frac{1}{5}\pi$	$\frac{1}{5}$	0·0460 0815
$\frac{1}{6}\pi$	$\frac{1}{6}$	0·0324 9137

THE SERIES $S_2(\alpha, u)$.

Let $$\varphi_n(u) = \frac{(1-u^{k-2})^2}{1-u^{2k}} \quad .. \quad .. \quad (15)$$

then the series $S_1(\alpha, u)$ can be written as follows :

$$S_2(\alpha, u) = \sum_{n=0}^{n_1-1}\frac{\varphi_n(u)}{k(k^2-4)^2} + Z_{n_1}, \quad .. \quad (16)$$

where the remainder

$$Z_{n_1} = \sum_{n=n_1}^{\infty}\frac{\varphi_n(u)}{k(k^2-4)^2} \quad .. \quad .. \quad (16_1)$$

One can see quite easily that for each

$$k > k_0 = \frac{\log_n\left(1+\frac{1}{u^4}\right) - \log_n 2}{-\log_n u} - 2 \quad .. \quad (17)$$

(or for each $n > n_0$, where $n_0 = -\frac{1}{2} + p_0$)

$$\varphi_n(u) < 1$$

The values of k_0 are given in Table 2.

TABLE II.

u	0	0·2	0·4	0·5	0·6	0·8	1
k	2	1·562	1·262	1·090	0·823	0·430	0

TABLE III.

$2a$	$\frac{u}{p} =$	0·2	0·4	0·5	0·6	0·8	1
2π	2	18·787	2·159	1·0028	0·5010	0·1206	0
π	1	1·8609	0·3029	0·1519	0·07974	0·02015	0
$\frac{2}{3}\pi$	$\frac{2}{3}$	0·3361	0·07926	0·04327	0·02399	0·006414	0
$\frac{1}{2}\pi$	$\frac{1}{2}$	0·08125	0·02709	0·01617	0·00950	0·002701	0
$\frac{1}{3}\pi$	$\frac{1}{3}$	0·008554	0·004839	0·003405	0·002256	0·000735	0
$\frac{1}{4}\pi$	$\frac{1}{4}$	0·001605	0·001230	0·0009850	0·0007276	0·0002739	0
$\frac{1}{5}\pi$	$\frac{1}{5}$	0·0004477	0·0003988	0·0003484	0·0002819	0·0001222	0

TABLE IV. $100\,\frac{J}{a^4}$

$2a$	$u=0$	0·2	0·3	0·4	0·5	0·6	0·7	0·8	1·0
2π	87·80	57·02	42·13	29·29	18·50	10·30	4·721	1·525	0
π	29·79	23·68	18·24	13·20	8·560	4·850	2·253	0·754	0
$\frac{2}{3}\pi$	14·43	12·76	10·67	8·173	3·260	3·053	1·482	0·487	0
$\frac{1}{2}\pi$	8·239	7·692	6·558	5·197	3·608	2·150	1·036	0·353	0
$\frac{1}{3}\pi$	3·492	3·401	3·123	2·647	1·972	1·253	0·642	0·225	0
$\frac{1}{4}\pi$	1·814	1·788	1·706	1·510	1·200	0·810	0·439	0·156	0
$\frac{1}{5}\pi$	1·064	1·055	1·025	0·932	0·776	0·554	0·318	0·118	0

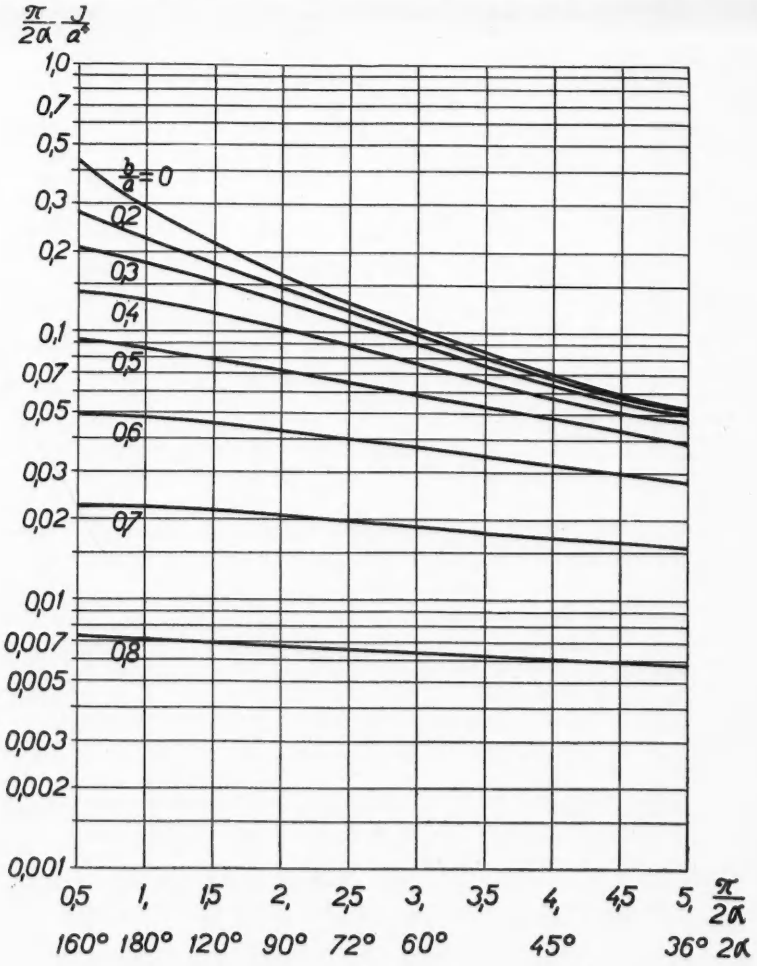


Fig. 2.

We assume that in equation (16_1) $n_1 > n_0$, then

$$Z_{n_1} = \sum_{n=n_1}^{\infty} \frac{\varphi_n(u)}{k(k^2-4)^2} < \sum_{n=n_1}^{\infty} \frac{1}{k(k^2-4)^2}$$

Hence it follows that

$$\sum_{n=0}^{n_1-1} \frac{\varphi_n(u)}{k(k^2-4)^2} < S_2(\alpha, u) < \sum_{n=0}^{n_1-1} \frac{\varphi_n(u)}{k(k^2-4)^2} + \Phi(\alpha, n_1), \quad \ldots \quad \ldots \quad \ldots \quad (18)$$

where $$\Phi(\alpha, n_1) = \sum_{n=n_1}^{\infty} \frac{1}{k(k^2-4)^2}. \quad \ldots \quad \ldots \quad (19)$$

Since $\lim_{n_1=\infty} \Phi = 0$ one can determine $S_2(\alpha, u)$ as accurately as may be required.

THE SERIES $\Phi(\alpha, n_1)$.

With the notation $k = \frac{1}{p}(2n+1)$ used above, the series (19) splits up as follows:

$$\Phi(\alpha, n_1) = \frac{p^2}{64} \sum_{n_1} \frac{1}{\left(n+\frac{1}{2}\right)\left(n+\frac{1}{2}+p\right)} - \frac{p^2}{64} \sum_{n_1} \frac{1}{\left(n+\frac{1}{2}\right)\left(n+\frac{1}{2}-p\right)} - \frac{p^2}{128} \sum_{n_1} \frac{1}{\left(n+\frac{1}{2}+p\right)^2} + \frac{p^2}{128} \sum_{n_1} \frac{1}{\left(n+\frac{1}{2}-p\right)^2}.$$

After the substitution $n = m + n_1 - 1$ and with equations (8_2) and (11), we get:

$$\Phi(\alpha, n_1) = \frac{p}{64}\left\{\psi\left(n_1-\frac{1}{2}-p\right) - 2\,\psi\left(n_1-\frac{1}{2}\right) + \psi\left(n_1-\frac{1}{2}+p\right) + \frac{p}{2}\left[\psi'\left(n_1-\frac{1}{2}-p\right) - \psi'\left(n_1-\frac{1}{2}+p\right)\right]\right\}. \quad \ldots \quad \ldots \quad (20)$$

THE SUM OF THE SERIES $S_2(\alpha, u)$.

According to the last two paragraphs we can determine the sum $S_2(\alpha, u)$ as accurately as may be required. The results of the corresponding computation are collected in Table III.

In computing this table we have taken into consideration that in the series (6_2) the term for $k = 2$ (i.e., for $p = n + \frac{1}{2}$) is an indeterminate expression whose value

$$\lim_{k\to} \frac{(1-u^{k-2})^2}{k(k^2-4)^2(1-u^{2n})} = \frac{(\log_n u)^2}{32(1-u^4)}.$$

THE TABLE OF $\frac{\mathcal{J}}{a^4}$.

From Tables I. and III. one finally obtains, by means of equation (7), the required dependence of the moment of inertia $\mathcal{J}$ on the outer radius a, the ratio $u = \frac{b}{a}$, and on the angle 2α of the annular ring-sector.

This dependence is shown in Table IV. and by the graphical reproduction in Fig. 2. The ordinates are plotted in logarithmic scale. For the sake of completeness the values for $u = 0{\cdot}3$ and $u = 0{\cdot}7$ have been determined by interpolation. In order to determine a special value of $\frac{\mathcal{J}}{a^4} = f(\alpha_1 u)$ for certain values of 2α, u not given in the table, we can use the following linear interpolation formula:

$$\left.\begin{aligned} f(\alpha, u) = {} & f(\alpha_1, u_1)\frac{\alpha_1(\alpha_2-\alpha)(u-u_2)}{\alpha(\alpha_2-\alpha_1)(u_1-u_2)} + \\ & + f(\alpha_1, u_2)\frac{\alpha_1(\alpha_2-\alpha)(u-u_1)}{\alpha(\alpha_2-\alpha_1)(u_2-u_1)} + \\ & + f(\alpha_2, u_1)\frac{\alpha_2(\alpha_1-\alpha)(u-u_2)}{\alpha(\alpha_1-\alpha_2)(u_1-u_2)} + \\ & + f(\alpha_2, u_2)\frac{\alpha_2(\alpha_1-\alpha)(u-u_1)}{\alpha(\alpha_1-\alpha_2)(u_2-u_1)} \end{aligned}\right\} \quad (21)$$

Here u_1, u_2 and α_1, α_2 are the values given in the table and nearest to the u, α values in question.

Note:

Table III. does not contain the values $S_2(\alpha, u)$ for $u = 0$, since in equation (7) the values of $u^4 S_2(\alpha, u)$ are used and

$$\lim_{u\to 0} u^4 S_2(\alpha, u) = 0$$

Example:

To find the moment of inertia $\mathcal{J}$ for an annular sector, with $a = 9$ cm., $b = 3$ cm., $2\alpha = 80$ deg. $= \frac{4}{9}\pi$

We determine

$$u = \frac{b}{a} = \frac{1}{3}$$

The nearest values of Table IV. are:

	$u_1 = 0{\cdot}3$	$u_2 = 0{\cdot}4$
$2\alpha_1 = \frac{1}{2}\pi$	6·558	5·197
$2\alpha_2 = \frac{1}{3}\pi$	3·123	2·647

According to equation (21)

$$\frac{100\,\mathcal{J}}{a^4} = 5{\cdot}319.$$

Therefore $\mathcal{J} = 0{\cdot}05319 a^4 = 349$ cm.^{4}, and the angle of twist per unit length

$$\vartheta = \frac{M}{349\,G}.$$

SELECTION OF SUITABLE STEELS FOR PUNCHES.

By J. A. HELLER and A. V. TURSINOV. (From *Stal*, No. 5/6, 1944, pp. 45-48, no illustrations.)

DIES for hot work are frequently made of 8 per cent tungsten steel or of steels containing nickel and molybdenum. Two other steels have recently been introduced which are considerably lower in cost. Their compositions are as follows :—

Steel ZIG60 : 0·3-0·4 C ; 0·4 Mn ; 0·4 Si ; 7·0-8·0 Cr ; 2·2-3·0 W.

Steel 5XGM : 0·5-0·6 C ; 1·0-1·5 Mn ; 0·4 Si ; 0·5-0·8 Cr ; 0·15-0·3 Mo.

It was found, however, that punches made with steel 5XGM show unduly rapid wear. Investigations were therefore carried out to ascertain the suitability of the following steels :

Steel Silchrome SX8 : 0·46 C ; 4·1 Si ; 0·45 Mn ; 8·18 Cr.

Steel 5XFM : 0·54 C ; 0·38 Si ; 1·39 Mn ; 0·72 Cr ; 0·28 Mo.

Steel Chromansil 35XGS : 0·35 C ; 1·4 Si ; 1·05 Mn ; 1·3 Cr.

Carbon steel No. 60 : 0·64 C ; 0·3 Si ; 0·7 Mn.

In the selection of these steels the following points were taken into special consideration :—

1. Steel of the Silchrome type has a high mechanical strength and good resistance against oxidation at high temperature.
2. Steel 5XGM had been recommended in technical literature as a suitable die steel for hot work.
3. Chromansil steel possesses high mechanical strength.
4. Carbon steel No. 60 is a low cost carbon steel.

The dies made of these steels were used for the punching of mild carbon steel at 1140-1160 deg. C.

The punching operation during which the die was in contact with the hot work piece took 5-8 seconds. The tools were provided with internal water cooling, the thickness of the cooled wall ranging from 20 to 30 mm. Surface cooling was not employed.

Heat treatment of the tools was carried out in accordance with standard practice as established for the steels in question (Table I).

After tempering the tools showed a troostitic microstructure, with the exception of the tools made of steel 35XGS and tempered at 650 deg. C., which had a sorbitic structure.

The wear of the tools manifested itself by "washing," that is, the appearance of erosion marks which had a depth of more than 0·4 mm. The appearance of these marks was accompanied by the formation of longitudinal fissures in the working surfaces of the tool, which eventually led to its breakdown.

Observations made during the working of the dies showed that upon the surfaces of those made of the steels 35XGS and 5XGM there appeared a number of scattered erosion marks which increased in size with further use of the tool. This "washing" of the tool occurred with particular rapidity in the case of the carbon steel No. 60. The steel SX8 showed a somewhat different behaviour insofar as on its surface a large number of shallow eroded spots were formed, which spread quickly over the surface during subsequent working of the tool.

The discarded dies made of the steels 35XGS and 5XGM had a bluish lustrous surface, while the tools made of the steels SX5 and No. 60 exhibited a dull surface of the type generally associated with the presence of scale. From an examination of the surface it was found that there existed a layer of scale which reached a thickness of 0·4-0·5 mm. after 100-150 operations of the die. In the case of the dies made of carbon steel No. 60, this scale layer had formed after only 40-70 operations. Immediately below this scale layer there existed a layer of oxidized metal of 0·7 0·9 mm. thickness containing a network of fissures easily visible to the naked eye. The dies made of steel 35XGS showed by far the best wearing qualities. Based on the wearing qualities of steel 5XGM the comparative tool lives were as follows :—

TABLE I.

Steel	Temperature of quench, deg. C.	Type of quench	Tempering temperature deg. C.	Brinell Hardness after tempering
SX8	1050	In oil to 250 deg. C., followed by cooling.	650	364-387
5XGM	825	Do.	540	364-418
35XGS	930	Do.	540	364-444
Do.	930	Do.	650	286-302
No. 60	850	In water to 150 deg. C., followed by air cooling.	380	340-480

Steel 5ZGM .. .. 100 per cent.
Steel 35XGS .. .. 200-220 per cent.
Steel SX8 .. .. 100-120 per cent.
Steel No. 60 .. .. 30-40 per cent.

The respective hardness values of the dies before and after use, as well as the critical points of the materials concerned, are given below.

TABLE II.

Steel	Brinell Hardness		Critical Point Ac_3 deg. C.
	New Tool	Broken-down Tool	
5XGM ..	364-418	444-550	790
35XGS ..	418-444*	196-241	860
Do. ..	286-387†	196-241	860
SX8	364-387	196-241	975
No. 60.. ..	340-380	196-241	760

* Tempered at 540 deg. C. † Tempered at 650 deg. C.

These data show that the hardness of steel 5XGM increases in operation, and it was found that the hardened case extended to a depth of 1·5-2·0 mm. The microstructure of this steel was fine-grained and showed fine martensitic acicular structure, while that of carbon steel No. 60 showed lamellar pearlite in the surface and fine-grained pearlite in the interior of the piece. Fine-grained pearlitic structure was also found to prevail in the broken-down dies of the steels 35XGS and SX8. It has this been proven by both micrographic inspection and hardness test that the contact of the die with the hot workpiece acts to increase the surface temperature of the tool to some 800-850 deg. C. But as Table II shows, this temperature is in excess of the Ac_3 point of the materials 5XGM and No. 60 ; while in the case of steels 35XGS and SX8 it still is below the critical point of the material. However, in the case of steel 5XGM the cooling effect produced by the internal water cooling of the die, was great enough to produce martensitic structure in the surface by quenching.

As regards the carbon steel No. 60, the cooling speed is below the critical rate required to produce hardening, but at the same time the surface temperature exceeds the Ac_3 point. It is for this reason that the surface strata of tools made of steels 5XGM and No. 60 with periodically repeated working sustain not only volume changes, caused by the temperature variations incurred in the working process, but also suffer additional volume changes due to the phase change which takes place in the material passing through the critical point. It is thus easily seen that the steel must fail because of thermal fatigue. Much smaller volume changes take place in the case of steel 35XGS since no phase change is involved. Dies made of this steel therefore have a much longer service life.

FRANCE

RATIONAL WORKING OF FURNACES FOR THE ELECTROLYSIS OF ALUMINIUM.

By L. FERRAND. (From *Journal du Four Electrique et des Industries Electrochimiques*, Vol. 54, No. 1, June, 1945, pp. 10-13. 1 illustration.)

A GENERAL theory of the working of electrolysis of fused electrolytes is described, by which the measurement of efficiency is brought down to a final measurement of voltages, if certain conditions are fulfilled, so that checks can be made as often as desired.

Let Δ be the current density in amps. (cm²),
K the density of heat losses in watts/cm²,
C consumption of energy D.C. per kg. of metal,
η efficiency,
α coefficient of internal resistance in ohms.

The four fundamental equations, which determine the working conditions of every electrolyzer of fused electrolytes are :—

$$\Delta = \sqrt{\frac{K}{\alpha}} \quad \text{in amps} \qquad (1)$$

$$K = (U - 2{\cdot}15\,\eta)\,\Delta \quad \text{in watts} \qquad (2)$$

$$U = \alpha\,\Delta + 2{\cdot}15\,\eta \quad \text{in volts} \qquad (3)$$

$$C = \frac{\epsilon U}{\eta} \quad \text{in Kwh} \qquad (4)$$

α is the ohm resistance of an imaginary conductor of variable section through which an elementary current, Δ, flows from positive to negative pole. This can be written in binomial form $\alpha = \frac{\lambda}{\mu\beta} + \omega$, where the first term represents the resistance of a small column of electrolyte of height λ (interpolar distance in cms) of mean section β cm² (1cm. back anode) β is therefore the coefficient of dispersion by which the anodic section must be multiplied to give the mean area of passage in the bath. μ is the specific conductivity of the bath (1·83 siemens per cm).

In practice the arithmetic mean of the area of the anode and of the hearth of the furnace is taken as the mean area of passage β, thus varies from 2·7 ohms for small furnaces to 1·25 ohms for large furnaces. It is a purely dimensional coefficient.

The 2nd term is the ohm resistance of a chain of simple conductors, equally traversed, in the current supply fittings, other than the bath, with a current Δ. It is thus a structural characteristic depending upon the method of construction of the hearth, the current densities, the anodes and their seals. It may vary, but only slightly, with wear of the hearth. It is 0·90 ohms for small furnaces and 1·1 ohms for very large ones.

It can thus be seen that α depends chiefly on the interpolar distance. The value of α is 2 ohms approximately for 10,000 amp. furnaces, 3 ohms for 30,000 to 50,000 amps. and 3·5 ohms for 100,000 to 125,000 amp. furnaces.

Equations (1), (2), (3) and (4) can be represented by a series of abacus with aligned points, which is extremely useful for the forecasting of certain characteristics of aluminium electrolysis, as well as for interpreting the various anomalies or mishaps in the running of the furnaces.

In Fig. 1 of these abacus, the left group (scale 1) refers to equation (1), the middle group (scale 3) to equation (3), scale Up for voltage loss $\Delta\ \alpha$ acts as return, and the right-hand group (scale 4) gives the equality (4).

PRACTICAL APPLICATIONS.

To determine K it is necessary to fix it by previous experiment for different working temperatures. During the time of observation, the temperature of the bath is held constant and so is the composition of the electrolyzing gas, so as to check the theoretical consumption of energy, by a thermo-dynamic calculation with final correction.

To find out the weight of metal cast, the depth of the metal in the bottom of the furnace is measured at the beginning and end of each period as well as the weight of aluminium added, to a constant amount if possible, at each anodic effect.

With U and I known by direct measurement, the mean output during the period is obtained by making a quotient of the energy consumed by the weight of metal produced, whence K is deduced by equation (2).

These observation periods should be repeated several times a year with zones of working temperature varying from —10 deg. C. to +30 deg. C, so as to establish a curve for K as a function of the working temperature. This graph, obviously peculiar to each type of electrolyzer, is then used by interpolation, and is an instrument for regulating operations which must be drawn up with care.

On the abacus have been shown the experimental results taken from three different prototypes of electrolysis furnaces built by the author during recent years, as shown on the following resumé :—

	30,000 A. Furnace (Oct.-Nov., 1942)		50,000 A. Furnace Apr. (1944) May	
	Cell 1.	Cell 2.		
Intensity, amps.	28,850	28,850	46,700	49,600
Interpolar distance tension, volts	4·435	4·6	4·43	4·5
Density of current	0·95	0·89	0·81	0·84
Output per cent	86·9	90	91·3	93·7
Consumption of energy, Kwh/t	15,220	15,240	14,450	14,300

(D.C. for electrolyzer only, without connection losses.)

It is rather remarkable that these working results are almost on straight lines, which shows that the general theory, as set out, gives a fairly good account of what takes place in practice.

Alignments.—For the 2-twin cells semi-clad, 30,000A, slightly differing, experimental readings give values for K :—

Cell 1 .. .. K=2·42 w/cm²
Cell 2 .. .. K=2·38 w/cm²

Scales 1. Taking Cell 1 as example, the value 2·42 for K is aligned with the value of the corresponding current density Δ = 0·95 A/cm², which gives an internal coefficient of resistance of 2·68 ohms as read on α.

Scales 3. Taking 2·68 on scale α (3) this is aligned with 0·95 on the oblique scale Δ ; point 6 thus obtained is joined on Up with the value of the working voltage on scale U (3), say, 4·45 volts, which gives a current efficiency of 87 per cent as against 86·9 per cent as measured.

Scales 4. Transferring the current efficiency to (4) and taking the voltage of 4·45 on U(4) this gives a consumption on C(4) at point 11, say, 15,250 Kwh, as against 15,220 as measured.

The same method is followed for the second cell, and it is now seen that what has been gained on voltage as compared with Cell 1 (4·45 volts instead of 4·605 volts) is lost in efficiency and the consumption is actually the same.

For the 50,000 A. furnace fully-clad, it was found in April, 1944 (cooling by natural convection) that—

U=4·43 volts, η=91·3 per cent, Δ=0·81 A/cm², α=3·03, and in May, with hearth cooling to increase losses artificially ;

U=4·5 volts, η=93·7 per cent, Δ=0·845 A/cm², α = 2·93.

To compensate for the cooling, the current density was increased by reducing the interpolar distance. Current consumption in both cases was less than 14,500 Kwh.

All the furnaces have four horizontal anodes, regulated by individual motors. A French electro-mechanical control desk provides automatic voltage control by suitable action on the motors.

PRACTICAL INTEREST OF THE ABACUS.

It is easy to complete the graduation of these scales so as to make them handier to use on electrolyzers of a given type by adding on the right of Δ, graduations as to function of intensity I for an anode of given section and to the density of α, a double graduation as a function of the interpolar distance λ.

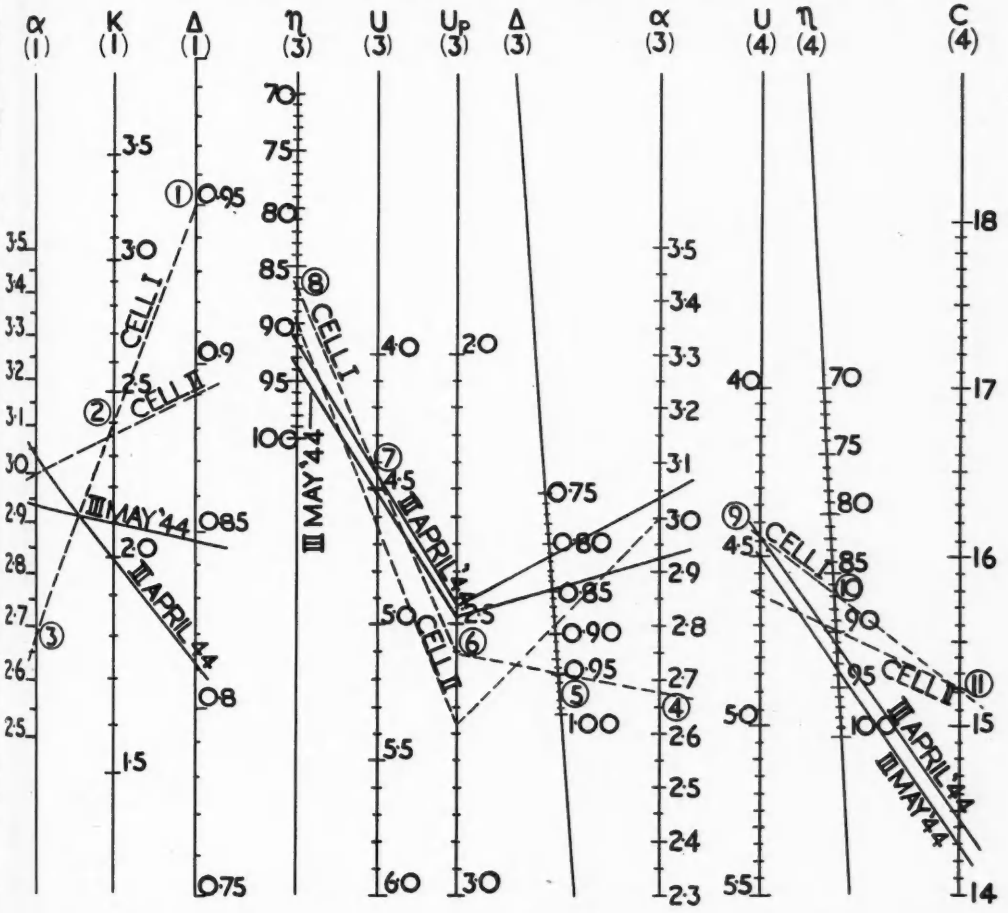
Here is an instance of the practical interest of this abacus :—

1. *Predetermination of output and consumption.*—U, I and K being known as functions of the working temperature on condition that the bath is stabilized at the working temperature, and also that the composition of the electrolyzing gas is the same as when K was determined, alignments are carried out as detailed above for Cell 1. In a few minutes the current efficiency can be obtained at point 8 and consumption at point 11, whereas otherwise monthly averages are all one has to work on. If efficiency and consumption are abnormal, immediate warning is given that the furnace is not working properly. The reason must then be discovered at once, be it faulty control of intensity distribution, formation of a mushroom short-circuit under an anode, and the remedy quickly applied.

2. *Changes in working temperature.*—A variation of 10 deg. C has an effect on the bath, and is sufficient to alter the output and increase anode consumption. Prolonged cooling can lead to an inversion of density.

The abacus also give new constants to be adopted, as follows :—Suppose that from the line 1, 2, 3, there is a lowering of temperature so that the density of losses K rises from 2·4 to 2·7. Two remedies are available to correct this :—

(*a*) Keep the intensity constant. The line pivots on 1 and passing through 2·7 on K cuts at point 3. It is sufficient to alter the interpolar distance shown by graduation λ on the right of scale α to obtain this increase of the coefficient of internal resistance from 2·68 to 3. There is a new value for U which is shown on the volt meter as soon as heat equilibrium of the bath is obtained. On Scale 3 line 4, 5, 6 pivots on 5 and passing through α = 3 cuts Up at a pivotal point which aligned with the voltage just read on the volt-



meter, gives a new value for η. On Scale 4 the two values of U and η give the amount of energy consumption by intersection with line C 4.

(*b*) Keep the interpolar distance constant. The functional line then pivots on 3 and passing through 2·7 on K cuts Δ about 1 A/cm², an increase of, say, 5 per cent in intensity. On scale 3, the line 4, 5, 6 pivots on 4, it pivots on axis Up at a point which, aligned with the new voltage value, cuts η at a point very near to the previous one, since λ has not been changed.

If working temperature increases making K change from 2·4 to 2·2, it is not possible to reduce the interpolar distance, which has presumably been reduced to a minimum, without affecting the output. Method (*b*) will then be followed by reducing the intensity.

3. *Variations in bath temperature.*—This situation, which may also arise from an internal fracture of the thermochemical balance, is treated in the same way as the previous trouble above.

4. *Variations in intensity at constant working temperatures.*—An accident in the generators, or a current failure, may force a temporary reduction of working intensity. In such a case, the line 1, 2, 3 pivots on 2 in a clockwise direction and, either, the interpolar distance can be increased, e.g., from α = 2·68 to α = 3·7 with Δ = 0·8 A/cm², or the intensity can be reduced by 15 per cent. Owing to the higher resulting voltage, as read on the voltmeter, new values of output and consumption will be obtained, or else, the same interpolar distance can be maintained which gives, by joining α = 2·60 to Δ = 0·8 A/cm², K = 1·72 W/cm². This means that to re-establish the heat equilibrium, extra heating power must be given by suitable means, equal to $(K_1 - K_2) = (2{\cdot}42 - 1{\cdot}72)$ S, either by 21 Kw. electrolyzer or 18,500 cal/H. This is a question for electrolyzers of variable power.

If it is desired to increase the density to absorb an excess of power, it is not possible to reduce the interpolar distance, and the excess power must be dissipated by losses through the cooling elements provided for this purpose.

5. *Variations in Internal Resistance.*—If this is increased owing to contact of bath and hearth, the functional line pivots on 2, but there can be no question of a measurable increase in the interpolar distance. If there is an isolated mishap, the reduced current density, corresponding to the increased internal resistance of only one electrolyzer, cannot be applied to the other electrolyzers. The bath should be cooled down, either artificially or by the addition of ingots of metal, whilst waiting for the cause of the trouble to be remedied.

If there is a drop in resistance, owing to the use of over-cooked anodes, it is sufficient either to increase the interpolar distance to keep α constant whilst dealing with isolated troubles, or to use a higher value for current density if the trouble is a general one.

These examples will explain the practical use of the abacus and how to follow constantly the working of a given furnace, to discover irregularities and to remedy them when the reason is known. It is an interesting technical method which allows of cutting out most of the empirical and chance methods which are used at present in the running of aluminium electrolyzers.

SWITZERLAND

CIRCULATING PUMP "MOBILA" FOR ACCELERATING THE FLOW OF HOT WATER IN CENTRAL HEATING INSTALLATIONS.

THE "Mobila" pump (Fig. 1) combines a rotary pump of the axial flow type, complete with its driving motor in the form of a single totally enclosed unit.

The impeller of the pump is formed in the interior of the motor rotor. The motor is of the "squirrel cage" type, capable of operation on either single or three-phase supply. The axis of the impeller lies in the flow path of the water, so that no stuffing boxes, or external connections are required.

The combined rotor and impeller (Fig. 2) is without slots or bars, and runs upon a shaft of hardened stainless steel, ground at the journals, and carried in bronze bushes of special composition, adapted for water lubrication, and capable of continuous silent operation.

The usual copper bars are replaced by copper plating the entire cylindrical surface of the rotor.

Fig. 1

The motor casing forms part of the pipe line, the stator winding being isolated from the rotor space, by a watertight bush of corrosion resistant material, which conforms to the magnetic requirements, and has a high resistance to rusting. The bush is rendered watertight by the two casing covers, special sealing devices being provided, so that the stator chamber remains dry.

The electrical characteristics of the motor are such that it can be switched "direct-on-line" of either single or three-phase supply, without the provision of any special starter, or condensers, and it is not affected by interruption in one phase of three-phase supply.

In the case of single phase supply, control can be provided by ordinary lamp switch.

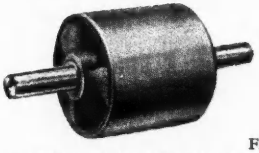


Fig. 2.

The pump will work in either horizontal or vertical position, and the simplicity of insertion in existing pipe lines is of particular advantage in fitting to existing central heating installations which suffer from defective circulation.

In general the following advantages are attainable by fitting this form of pump :—

1. Total absence of noise, by the use of rotor without slots or bars.
2. Ideal simplicity of fitting into pipe line, there being neither base or angle connections, by reason of the axial flow through the rotor.
3. Elimination of stuffing boxes and all external connections requiring attention.
4. Little resistance to flow when motor is not running.
5. Adaptability to operation on either single or three phase supply.
6. For single phase working, no condenser is necessary, hence centrifugal starters, relays, or other devices are eliminated.
7. First cost is moderate, owing to simple construction, small dimensions and light weight.

PRODUCTION OF ELECTRIC POWER FROM PEAT.

By L. MALM. (From *Jernkontorets Annaler*, Vol. 129, No. 3, 1945, pp. 123-161, 19 illustrations.)

DRY peat has a chemical composition which largely corresponds to that of wood, the average composition being as follows :

C	= 50	per cent
H_2	= 6·0	,,
O_2	= 42·2	,,
N_2	= 0·6	,,
S	= 0·2	,,
Ash	= 1·0	,,
Total	100·0	,,

The upper calorific value of dry peat is 4,870 kcal per kg., and the lower calorific value is 4,550 kcal per kg. The lower calorific value of wet peat can be expressed as function of the moisture content by the relationship :

$$W_{eff} = 4{,}550 - 5{,}135 \cdot F \text{ kcal/kg.} \quad .. \quad (1)$$

where F is the moisture content. In connection with the calculations given below, it is convenient to establish the combustion data on the basis of 1 kg. of dry peat. Fig. 1 shows the composition of the products of combustion in mol per kg. of dry substance and the calorific value in kcal per kg. for different moisture contents. It is also seen that the constituents CO_2 and SO_2 = 41·7 mol per kg. are independent of the moisture content. The same applies to the constituents $O_2 + N_2$ which vary with the CO_2 content (measured for the dry gas). As Fig. 1 shows, it is $O_2 + N_2$ = 255·4 mol per kg. if the CO_2 content is 14 per cent, and $O_2 + N_2$ = 278·4 mol per kg. for a CO_2 of 13 per cent. The only constituent of the combustion products which is affected by the moisture content is the water vapour. Referring to Fig. 1, it is seen that the water vapour percentage increases slowly at first with the moisture content of the peat, but rises rapidly as a moisture content of 70 per cent is approached. The rapid decrease in the effective calorific value of the peat with increasing moisture content explains the need for reducing the moisture percentage by drying as far as possible. An increasing moisture content of the fuel must necessarily result in an increase in the quantity of products of combustion, because of the increased amount of water vapour contained in the latter. This relationship is shown in Fig. 2.

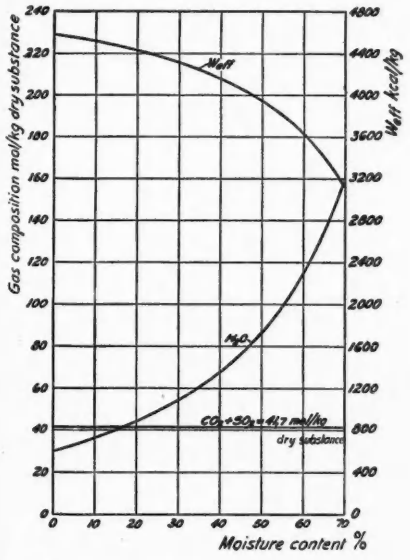


Fig. 1. **Composition of furnace gases and lower calorific value *Weff* in kcal. per kg. dry substance for varying moisture content.**

For CO_2 = 14 per cent, O_2+N_2 = 255·4 mol/kg.
For CO_2 = 13 per cent, O_2+N_2 = 278·4 mol/kg.

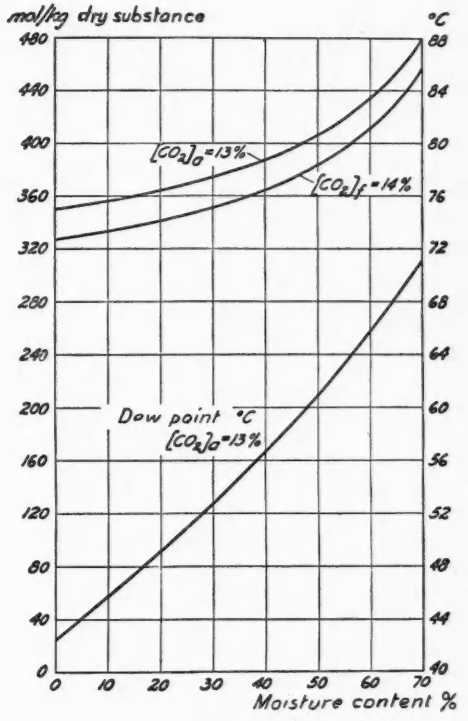


Fig. 2. **Quantity of furnace gases for CO_2 = 14 per cent in furnace and 13 per cent in boiler.**

Pre-drying of the peat is particularly important from the aspect of safeguarding complete combustion. The diagram given in Fig. 3 shows the enthalpy of the products of combustion as function of the gas temperature for different moisture percentages, the curve A — B indicating the theoretical temperatures of combustion without pre-heating of the combustion air. Corresponding curves for air-preheats ranging from 100 deg. C. to 400 deg. C. are also charted. From these curves it would appear that combustion temperatures in excess of 1000 deg. C. can be achieved by burning peat of 70 per cent moisture with an air preheat of 400 deg. C. Actual experience, however, has shown that this is not feasible. For a moisture content of 60 per cent the graph shows a theoretical combustion temperature of 1050 deg. C. if no

air preheat is used, and of 1260 deg. C. if the air is preheated to 400 deg. C. But here again practical experiences have shown that satisfactory combustion cannot be maintained without the employment of air preheat, and that 300–400 deg. C. preheat is essential. Satisfactory combustion, even without any preheating of the air, can, however, be achieved if the moisture content is reduced below 50 per cent. Fig. 3 also shows that there exists a lower limit in the neighbourhood of 30 per cent below which it is useless to reduce the moisture content.

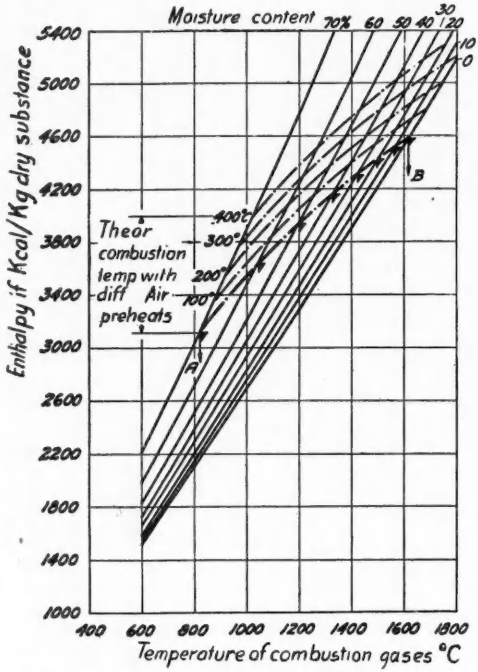


Fig. 3. **Enthalpy of furnace gases in furnace for CO_2 = 14 per cent.**

The influence of the moisture content upon the stack loss of a steam boiler is charted in Fig. 4. This graph also shows the increase in the dew point with the moisture content. As the gas temperature at the stack must be kept at a certain level above the dew point, the adverse effect of high moisture content upon boiler efficiency is obvious. Moreover, the loss due to combustible in ash also increases with the moisture content, especially if the moisture content exceeds 60 per cent. It is for these reasons that a reduction of the moisture content by drying to 30–50 per cent must be considered essential. This drying process can be carried out either by low pressure steam or by flue gas.

DRYING BY MEANS OF LOW PRESSURE STEAM.

A typical steam circuit of a plant in which the peat is dried by means of low pressure steam is given in Fig. 5. From the boiler *AP* which is built for a steam pressure *p* kg. per sq. cm. gauge, steam is supplied at the pressure 0·85 *p* kg. per sq. cm. gauge and the temperature *t* deg. C. to 2 turbines, of which the unit *MT* is a back-pressure

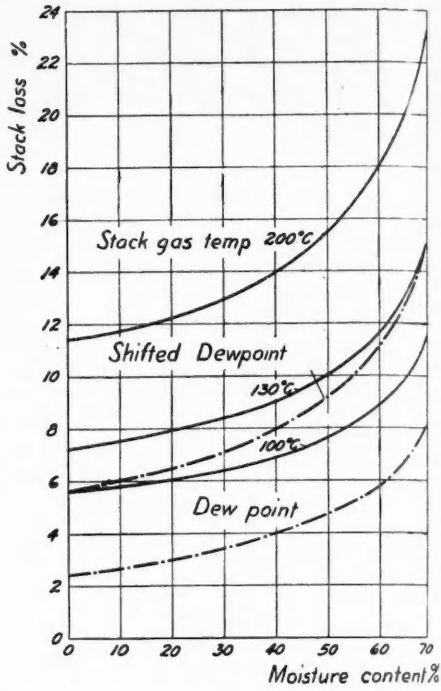


Fig. 4. **Stack loss in per cent of lower calorific value of dry peat.**

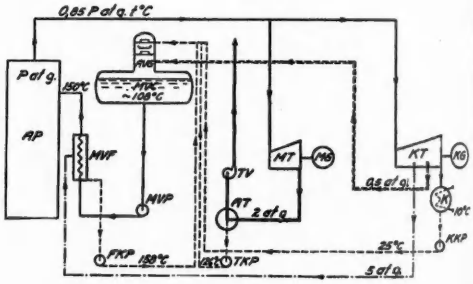


Fig. 5. **Steam circuit of plant with steam-drying of peat.**

machine driving the generator *MG*, and the unit *KT* is a condensing machine coupled with the generator *KG*. Steam at 2 kg. per sq. cm. gauge pressure is delivered by the back pressure turbine *MT* to a battery of steam dryers *AT*, from which the drying air laden with water vapour is discharged by means of the dryer fan *TV*. The condensate from the steam dryer is delivered by the centrifugal pump *TKP* to the deaerator *AVG* of the storage vessel *MVC*. The condensate of the condenser *K* is also discharged into this deaerator by means of the centrifugal pump *KKP*. The condensing turbine *KT* is provided with two extraction stages, one at 0·5 kg. per sq. cm. g. pressure for preheating the feedwater to 108 deg. C. in the storage reservoir, and another at 5 kg. per sq. cm. g. for heating the feedwater to 150 deg. C. in the closed heater *MVF*, from whence it is passed into the economizer. A centrifugal pump

FKP is provided to deliver the heating steam condensate from the feedheater *MVF* to the deaerator *AVG*.

In calculating the amount of dry peat in kg. required per Kw. hr. the following notations will be employed:

Amount of dry matter delivered to dryer	G kg. per hr.
Moisture in peat as supplied	F, $0<F<1$
Moisture in peat as dried	F_x, $0<F_x<1$
Steam consumed in dryer	α kg. per kg. H_2O
Steam consumption of back pressure turbine	m kg., per kWh.
Effective calorific value of peat with F_x moisture	W_{eff} kcal. per kg.
Boiler efficiency	η_p, $0<\eta_p<1$
Enthalpy of steam at turbine throttle	i_1 kcal. per kg.
Enthalpy of condensate after dryer	i_t kcal. per kg.
Enthalpy of turbine condensate	i_k kcal. per kg.
Steam rate of condensing unit	k kg. per kW. hr.

The output of the back pressure turbine is given by:

$$N_m = \frac{\alpha \cdot G}{m}\left[\frac{F}{1-F} - \frac{F_x}{1-F_x}\right] \text{ kW.} \quad .. \quad (2)$$

The fuel required for back pressure power and drying amounts to:

$$B_m = \frac{\alpha \cdot G\,(i_1-i_t)}{W_{eff} \cdot \eta_p}\left[\frac{F}{1-F} - \frac{F_x}{1-F_x}\right] \text{ kg. per hr.} \quad (3)$$

The fuel required for the power developed by the condensing unit is:

$$B_k = (G - B_m) \text{ kg. per hour}$$

and according to equation (3)

$$B_k = G\left[1 - \frac{\alpha\,(i_1-i_t)}{W_{eff} \cdot \eta_p}\left(\frac{F}{1-F} - \frac{F_x}{1-F_x}\right)\right] \text{ kg. per hr.} \quad (4)$$

The output of the condensing unit is given by:

$$N_k = \frac{G \cdot W_{eff}\ \eta_p}{(i_1-i_k)\ k}\left[1 - \frac{\alpha\,(i_1-i_t)}{W_{eff} \cdot \eta_p}\left(\frac{F}{1-F} - \frac{F_x}{1-F_x}\right)\right] \text{ kW} \quad .. \quad (5)$$

and the total output is therefore:

$$N = N_m + N_k$$

Thus

$$\frac{N}{G} = \frac{W_{eff} \cdot \eta_p}{(i_1-i_t)\ k} - \left(\frac{F}{1-F} - \frac{F_x}{1-F_x}\right)\left(\frac{i_1-i_t}{i_1-i_k}\frac{\alpha}{K} - \frac{\alpha}{m}\right) \text{kWh. per kg.} \quad .. \quad (6)$$

If the peat is supplied to the plant with sufficiently high moisture content F_o, the total output N kW will be equal to the back pressure output N_m. With $N_k = 0$ it will then be

$$W_{eff}\ \eta_p = \alpha\,(i_1-i_t)\left(\frac{F_o}{1-F_o} - \frac{F_x}{1-F_x}\right)$$

or

$$F_o = \frac{\dfrac{F_x}{1-F_x} + \dfrac{W_{eff}\ \eta_p}{\alpha(i_1-i_t)}}{1 + \dfrac{F_x}{1-F_x} + \dfrac{W_{eff}\ \eta_p}{\alpha(i_1-i_t)}} \quad .. \quad (7)$$

Either formula (6) or formula (7) may be used to compute the fuel consumption, providing the factor α is known. The latter may be assumed to equal $\alpha = 1{\cdot}5$ kg. per kg. H_2O. With the use of formula (6) the following values are obtained:

Moisture content F_x after drying ..	0	30	40	50	60	70	per cent
Consumption peat for steam pressure of:							
50 kg. per sq. cm. g.	0·897	0·942	0·974	1·006	1·068	1·312	kg. per kW. hr.
80 ,, ,,	0·845	0·886	0·911	0·956	1·025	1·280	,,
110 ,, ,,	0·824	0·861	0·895	0·932	0·998	1·260	,,

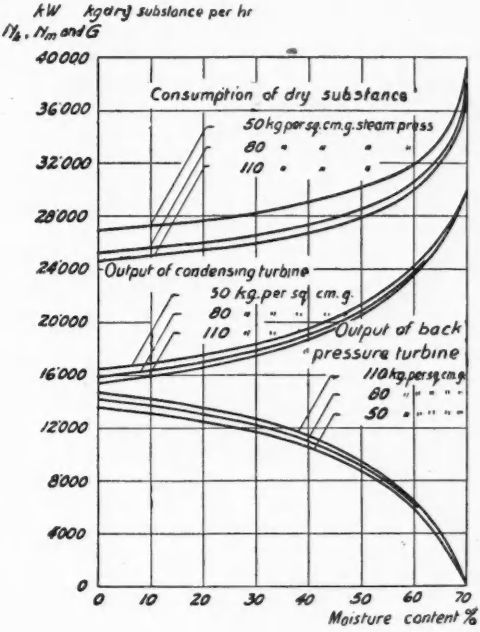


Fig. 6. Peat consumption of 30,000 kW. plant burning pressed peat of 70 per cent moisture content with varying final moisture content.

Fig. 6 indicates the amount of peat in kg. (dry basis) per hour required for a 30,000 kW. plant as function of the moisture content. The power produced in back pressure and condensing operation respectively is also given.

The sectional elevation of a power station based on the utilization of peat with 70 per cent moisture content is shown in Fig. 7. Here the pressed peat is delivered by a conveyor *VTB* to the steam dryer *AT* which is housed in a separate building. By means of a conveyor system *TTB* the dried peat is delivered to the boiler house. The dryer-air laden with water vapour is exhausted by means of a fan *TV* which passes the air into the cyclone separator *TC*. The particles retained in the cyclone are returned to the conveyor belt *TTB via* a rotary sluice *CHJ* and a screw conveyor *ST*.

In the boiler house the peat is removed from the conveyor belt *TTB* by means of the elevator, which discharges the peat into the overhead bunker *TB*, which is equipped with a feeder screw at its lowest point. This feeder screw discharges the peat into the shaft dryer *TS* which is in communication with the boiler furnace proper. This shaft is swept by a mixture composed of cold air, of air preheated to 400 deg. C., and of hot furnace gases. From the bottom of the shaft dryer the peat is passed on to the mill, which, in turn,

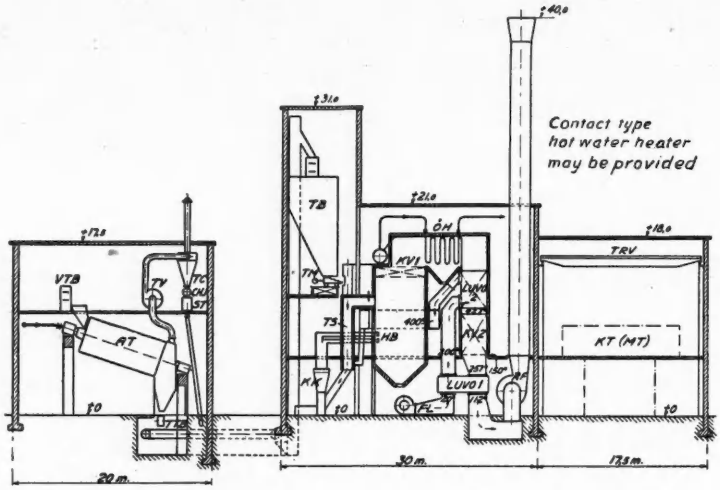


Fig. 7. **Proposed steam plant with steam dryer for handling peat of 70 per cent moisture content.**

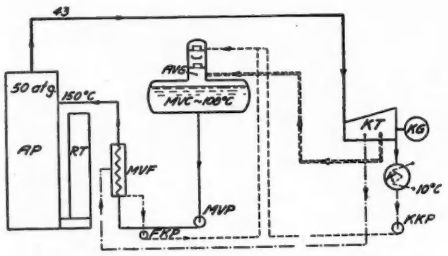


Fig. 8. **Plant circuit for steam plant with flue gas drying of peat.**

supplies the corner burners *HB* of the boiler furnace. The furnace will preferably be equipped with two burners per corner, that is, with eight burners, to be supplied by two mills. Upon leaving the furnace, the products of combustion pass over the evaporating surface *KV*1, the superheater *OH*, the high temperature air preheater *LUVO*2, the economizer *KV*2, and finally over the low temperature air preheater *LUVO*1, into the induced draught fan *RF*, and hence to the stack.

DRYING BY MEANS OF FLUE GAS.

Fig. 8 shows the basic circuit of a plant equipped for the flue gas drying of the peat fuel. It will be noted that this plant contains neither back pressure turbine nor steam dryer; while the boiler is equipped with the flue gas dryer *RT* capable of handling peat of 70 per cent moisture content. There will be a considerable reduction in the flue gas temperature due to the heat absorption in the dryer, and this fact will have to be taken into account when determining the convection surfaces of the boilers. The gas temperature required at the inlet of the dryer as function of the initial moisture content of the peat is given in Fig. 9.

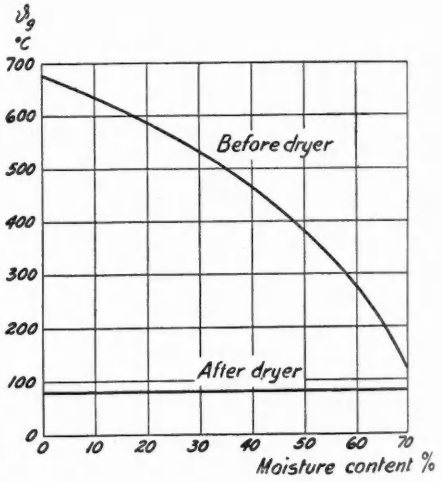


Fig. 9. **Temperature of flue gases before and after peat dryer.**

The estimated fuel consumption in kg. (on the dry basis) per kW. hr. is as follows:

Moisture content F_x of peat after drying	0	30	40	50	60	70	per cent
Fuel consumption at steam pressure of:							
50 kg. per sq. cm. g. ..	1·09	1·095	1·105	1·12	1·14	—	kg. per kW. hr.
80 ,, ,, ...	1·06	1·065	1·075	1·09	1·111	—	,,
110 ,, ,, ...	1·04	1·045	1·06	1·07	1·09	—	,,

A comparison of the two dryer systems shows that with an initial moisture content of 70 per cent, and a final moisture content of the peat of $F_x = 30$ per cent, the fuel consumption of the 80 kg. per sq. cm. g. plant equipped with flue gas dryer will be some 20 per cent above that of the plant equipped with steam dryer.

CRUSH TRUING OF GRINDING WHEELS FOR FORM TOOL GRINDING.

A symposium by C. J. WILSON, L. B. BELLAMY, L. A. KITCHEN, W. MORELAND and G. SKOG. (From *The Tool Engineer*, Vol. 15, No. 2, September, 1945, pp. 18-28, 25 illustrations.)

THIS symposium consists of five independent articles, as a result of which there is naturally some overlapping and repetition. The advantages of crush truing are well presented, and in some detail, and in addition certain disadvantages are mentioned.

SCOPE OF CRUSH-TRUING.

The basis of crush truing is repetition work of profiled work-pieces from mass production down to, say, 20 work-pieces; either form-grinding on a profile grinder or profile-truing of the wheel by means of diamond has to be resorted to for smaller batch numbers and single pieces. The profiling of grinding wheels for small quantities by means of a universal diamond truing device, is relatively expensive, not only in set-up, but also in diamond consumption and wheel consumption. If the roller is not produced on a profile grinder, for instance, when such an expensive machine is not available for the job, the roller itself has to be produced by this method.

MATERIAL OF ROLLER.

Unhardened mild steel crushers have frequently been used (Kitchen, Wilson), but their life is naturally much shorter than that of hardened rolls (Bellamy). The unhardened roller has the advantage of being turned out quickly on the lathe, and is not subject to hardness distortion. High-speed steel GM, F1 or M-2 hardened to Rockwell C 64-66 are suggested by Kitchen and H.S.S. 18-4-1 of Rockwell hardness C 65 to 67 by Wilson. Material GM-46 M tool steel was also found to be very good, and is easier to grind than H.S.S. (Kitchen). Nitriding shows some promise, and very good results have been obtained from tools of cast high-speed steel (Moreland). Sintered carbide rollers are suggested (Bellamy, Moreland) or at least sintered carbide inserts (Kitchen).

MOUNTING OF THE ROLLER.

Crushers are preferably mounted in some type of holding fixture, usually on antifriction bearings to allow free rotation and minimise end play (Bellamy); Kitchen recommends enclosed Timken roller bearings, but also suggests placing the spindles of the roller for surface grinders in open V-blocks or open U-brackets (cradles), which provide only a half-bearing; this gives the possibility of rapid exchanges as the grinding wheel pressure only acts downwards (Fig. 1). A necessary requirement is that wheel spindle and crusher spindle should be absolutely parallel (Bellamy) and this is effected by locating the device with tongues to fit the table slot (Skog). This exposes the device to the abrasive dust coming from the wheel and a good alternative is to mount it on the wheel head and so keep away swarf and coolant (Skog). Kitchen suggests welded fixtures (Fig. 2).

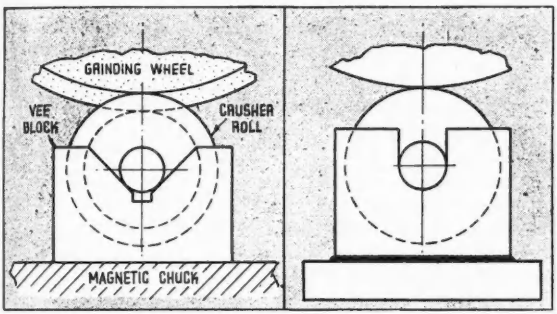


Fig. 1. **Mounting of the roller in open cradles.**

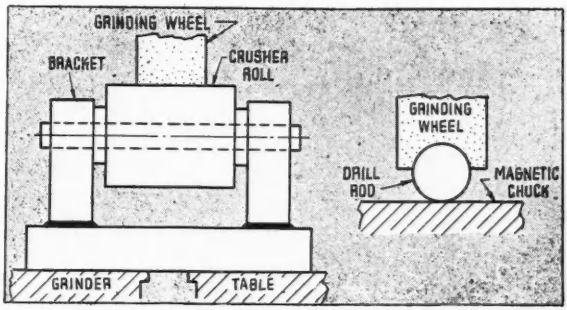


Fig. 2. **Simple supporting bracket for crusher roller and use of drill rod as crush-dressing device.**

Abstract by courtesy of the Diamond Research Department, 32/34, Holborn Viaduct, London, E.C.1.

INEXPENSIVE ROLLERS — KEEPING THE FORM.

Some engineers believe that the crusher costs are usually too high, and in the following, some of the suggestions made in the symposium are compiled to show how costs can be reduced.

In cutter grinding operations, roller costs can be reduced considerably by using worn-out milling cutters as blanks and grinding the form on them (form grinding of roller, see under Scope of Crush Truing). It is strongly recommended to keep one roller as the master (Moreland), and to produce a set of three or more service rollers cheaply by crushing the form of a surface grinder on the wheel with the first roller as master, and to grind and re-grind rollers when they lose form.

In another instance, the same roller was provided with three grooves side by side. The first groove is used for roughing, the second for semi-finishing, and the third for finishing (Kitchen). This saves time and guarantees long life.

A new method, suggested for surface grinders, is crushing with a piece of round steel rod of a diameter suitable for producing the desired radius. The rod is held on a magnetic chuck, and the grinding wheel brought down to the rod until the desired radius is crushed into the wheel. The circumference of the grinding wheel should not exceed the length of the chuck, which is fed in a horizontal direction (Fig. 2). The same method seems to be applicable, of course, to forms other than round ones.

GRINDING WHEEL PROBLEMS.

Bellamy recommends that the following factors should be considered in wheel selection: (1) Kind of material to be ground. (2) Amount of stock to be removed. (3) Gauging limits and surface finish required. (4) Area of contact between wheel and work. (5) Type of grinding machine and its condition. (6) Work speed. (7) Wheel speed. (8) Type of operation. He further recommends 7 to 12 in. diam. wheels with 80 to 150 grit, grades I and K. Finer grain and tighter structures are generally used for crush grinding and surface grinding machines. For external diameter and plunge grinding, 120 to 220 grit and grades K to N are preferred. Better grinding action is said to be obtained with regular grains than with more highly refined or premium grain wheels. Tougher clay bonds enable the wheel to retain its form for a longer period than some of the more pliable vitreous (glass) bonds. Kitchen recommends starting with the (vitrified bond) wheel just on the machine and to watch the finish; one may, perhaps, change, later on, to a slightly softer grade and to a finer grain size. Moreland found that wheels taken off the spindle and put on again needed redressing owing to varying hole sizes and wheels being not perfectly balanced; this increases wheel costs and crusher wear. He suggests better balanced wheels, i.e., wheels which remain in balance regardless of the number of redressing operations. The bore of the wheel should be maintained within 0·001 in. for size and roundness, the commercial standard being 0·002 to 0·005 in., but many wheels measured plus 0·008 to 0·010 in. All wheels for crush truing should have a bushing or be sprayed with lacquer in the bore. Steel inserts are better than the usual lead inserts (Moreland, Bellamy), since they reduce set-up time, wheel and crusher wear; the grinding wheel can be removed and replaced without re-crushing.

CRUSHING PROCEDURE.

Various crushing procedures are in use. Two main methods are: either to drive the crushing roller or the grinding wheel at a speed very much less than the usual grinding speed. With both methods, very satisfactory results have been obtained (Wilson). Driving of the grinding wheel is recommended for very wide contours and heavy spindles (Wilson). The simplest method is to rotate the roller by a hand-operated crank. Peripheral speeds should be 150 to 250 ft./min. (Bellamy), or 225 ft./min. (Kitchen), and 300 ft./min. (Wilson), the latter data applying to a roller to $2\frac{3}{8}$ in. wide and 5 in. outside diameter. Motor driven crusher roller drives have been developed by the Sheffield Corporation.[1] Usually, geared motor units of $\frac{1}{4}$ h.p. (Wilson) to $\frac{1}{3}$ h.p. (Skog) are used. In spite of this, it seems to be preferred for the grinding wheel to drive the roller. When the grinding wheel is driven the machine should have a two-speed spindle, and the low speed should give about 60 to 80 ft./min. (Moreland). Speed changes without manual operation are required. In some instances the wheel spindle also seems to have been "cranked" (Skog), but this does not appear to be advisable. According to Kitchen there is no rule for the ratio of crusher diameter to wheel diameter as this depends on the conditions of the job. Satisfactory combinations were:

Crusher diameter ..	in.	$\frac{5}{8}$	7	4
Wheel diameter ..	in.	14	30	$3\frac{3}{4}$
Ratio ..		1/22	1/43	1/0·94

GENERAL REQUIREMENTS.

The outcome of this symposium seems to be that crusher truing is not merely a method which can be applied in every instance, but that it needs what is called in America, "engineering," i.e., careful consideration, development work and special devices. In particular new and stringent demands are set up in connection with the machine. As is already known[2] the low speed and large contact area put an extra heavy load onto the wheel spindle. Comparatively few machines have rigid enough spindle construction (Moreland, Kitchen); bigger spindle diameters are particularly advocated, as well as wider wheels, which should be at least 2 in. As suggested by Zikel[3] in England, for cutter grinding machines it should be possible to place the roller into the female centres. (Moreland.)

COSTS AND PRODUCTION TIMES.

In some circles, it is believed that crusher cost is too high to offset other savings, since set-up and truing times may be longer than in diamond truing. Moreland presents the following data relating to a certain tool from the Chrysler Jefferson Plant with data relating to another tool in brackets. Lots of 20 tools formerly had to undergo 4 (3) operations, total grinding time per tool 30 (20) min. With crush trued profile wheels this time is reduced to 2 ($1\frac{1}{4}$) min. Average set-up and wheel truing time, 40 (30) per cent less than with the old method. With a saving of $\frac{1}{2}$ hour per tool, 10 hours are saved for 20 tools. If the roller was worn out after 10 truings, which is considered ridiculously low, the total saving would be 100 hours. The crusher could be made in less than 20 hours and pay for itself on 40 (60) tools. Savings over up to 40 min. per tools have been obtained.

Crushers have been used satisfactorily over 50 times and even up to 120 times in corner rounding (Moreland), but this is obviously a question of the material used. Average roller life according to Kitchen is 50 to 80 crushings.

REFERENCES.

[1] *Machinery* (New York), Vol. 51, 1945, p. 202.
Machinist (Lond.), Vol. 89, 1945, p. 651 (Aug. 18).
[2] Pulcrush Form Grinding, *Industrial Diamond Review*, Vol. 5, 1945, p. 81.
[3] W. Zikel, Crushing Wheels for Form Grinding Without Special Equipment, *Machinery* (Lond.), Vol. 66, 1945, pp. 61, 67 (Jan. 18).

TECHNICAL PROBLEMS IN DESIGN AND OPERATION OF AIRCRAFT OF LARGE STOWAGE SPACE.

By R. GROCHALSKI. (From *Luftwissen*, Vol. 10, 1943, pp. 301-309.)

THE principal task in the design of large stowage space aircraft rests with the provision of sufficient range. The greatest distances to be covered by non-stop flights are those between America and Europe, i.e., some 6500 km. Taking a reasonable head wind into account, the required range is about 8000 km. Thus, with an average cruising speed of 400 km/hr. the time of crossing is 20 hours.

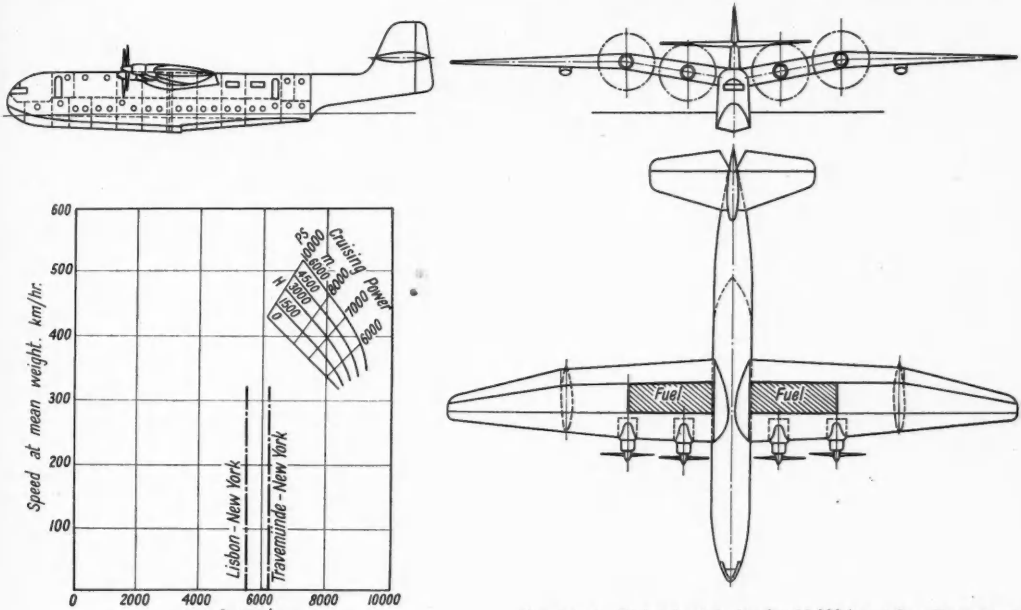


Fig. 4. **Speed-range diagram for design. Figs. 1-3.** Take-off weight, 71 tons. Take-off power, 14000 h.p. Wing area, 240 m². Fuel, 27 tons. Specific Fuel consumption, 190 gr. per h.p. hr. Aspect ratio of wing, A = 10.

Figs. 1 to 3. **Flying boat design for 14,000 h.p. take-off power and 70 tons all-up weight.**

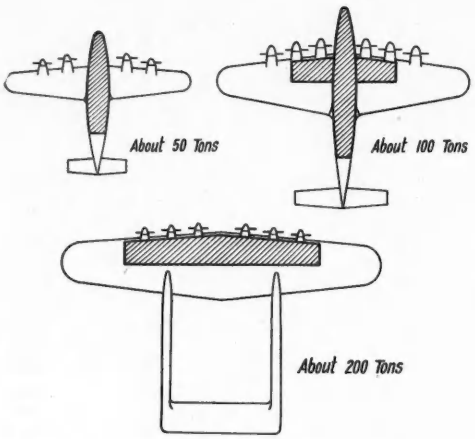
POSSIBILITIES OF HIGH ALTITUDE FLIGHT.

The required high speed is possible only at high altitudes.* Investigations into the weather showed that in the region of the troposphere, i.e., up to approximately 11 or 12 km, bad weather conditions must be reckoned with. This fact alone can make void the advantages of high altitude flying. Flying in the stratosphere, i.e., above 12 km altitude, still presents formidable technical difficulties. The pressurizing of the cabin, turbo-supercharging and other equipment required for high altitudes, increase considerably the weight of the aircraft. For instance, at 9 km altitude, the increase in engine weight is about 25 per cent, and the increase in fuselage weight is estimated to be as high as 50 per cent.

MINIMUM SIZE OF LONG-RANGE AIRCRAFT.

In order to design a long-range aircraft, the requirements of the operating companies must be known. American companies specify a range of 8000 km, and an average cruising speed of 400 km/hr. Accommodation is to be provided for 50 passengers, and freight to be carried of the order of 2000 to 3000 kg. Flying boats like the Dornier Do 26 and the Short Empire, of between 16 and 20 tons, cannot even approximately fulfil these requirements. The minimum size satisfying the above requirements will be from 75 to 100 tons. With increasing size, Consolidated, stepping up from a 50 ton model to a 76 ton model, increased the wing loading and reduced the power loading. The required high wing loading is an important criterion to the question, " landplane or flying boat ? " and can be decisive. To achieve a great range by economical means, the aircraft must fly near its optimum gliding angle, i.e., at high values of the lift coefficient. This leads either to high altitude flying or to small wing areas. With land planes, the increase in wing loading is limited by the length of the take-off run that can be made from available airfields. No such restrictions exist with the flying boat. Moreover, after unsticking there is no specified clearance for the flying boat as there are no obstacles and, therefore, the rate of climb of the flying boat can be kept as

* The paper obviously refers to the use of the conventional internal combustion engine. The author's views should altogether be regarded as obsolete by now in view of the fact that the B.O.A.C. is considering the project of Saunders-Roe Ltd. of a 120 ton flying boat to be driven by six turbo jet-propulsion engines driving contra rotating propellers. As, however, this flying boat will take three years to be completed, and the present controversy, flying boat (British conception) versus land plane (American conception) as applied to trans-oceanic service is by no means settled, we thought it of interest to publish an abstract of this article, written three years ago. It applies rather to conditions existing to-day. In another three years, when the present project matures, the speed will be 300 miles per hour, range 5000 miles. Flying up to 35,000 ft., the 3,450 miles non-stop flight between London and New York will take about 12 hours flying; passengers carried number 100.—Ed., E.D.



Figs. 5 to 7. Possibilities of stowage for various sizes of aircraft.

low as desired until the most economical cruising speed is reached. The larger the flying boat the greater the improvements in space condition. Figs. 1 to 3 show a 70 ton flying boat design. For take-off a power of 14,000 h.p. is provided. The wing loading is 300 kg/m², power loading 5 kg/h.p. From the range diagram, Fig. 4, a cruising speed of 430 km/hr, and a range of 8700 km can be read off. With engines throttled, over 9000 km can be reached. Space is provided for 48 passengers.

POSSIBILITIES OF THE ALL-WING AIRPLANE.

The question of whether the all-wing airplane is the suitable type for large stowage space is also dependent on the absolute size of the aircraft. From the point of view that favors increasing wing loading, the all-wing airplane does not seem to be of promise, as its wing area decreases relatively. Figs. 5 to 7 show various possibilities of stowage with increasing size of aircraft; only from 200 tons upwards will it be possible to have all stowage space in the wing.†

†This view was held by Dr. H. Roxbee-Cox in his 1940 Wilbur Wright lecture, and to those further interested in the subject, we cannot do better than refer to this lecture giving *inter alia* 80 references to literature (see *the Journal of the R.Ae.S.*, Sept., 1940, pp. 681-754, "Looking Forward. Prolegomena for a Detailed Study of the Future of British Civil Aviation").—Ed., E.D.

SWEDEN

PRESSURE-OIL OPERATED AUXILIARY MACHINERY IN STEEL PLANTS.

By M. OLEDAL. (From *Teknisk Tidskrift*, Vol. 75, No. 42, October 20th, 1945, pp. 1139-1144, 4 illustrations.)

THE control of mechanisms by means of pressure-oil offers the advantages that—

(1) wear is considerably reduced, as the working component parts are largely submerged in oil;
(2) the risk of breakdowns is reduced, as the maximum force exerted can be limited to a predetermined value by the corresponding setting of relief valves;
(3) the operation is smooth and gradual, and accelerations as well as decellerations at commencement and termination respectively of the working cycle can easily be adjusted;
(4) the choice of appropriate piston dimensions makes it possible to obtain different speeds for the forward and the return stroke;
(5) various movements can be executed in sequence by automatic coupling mechanisms.

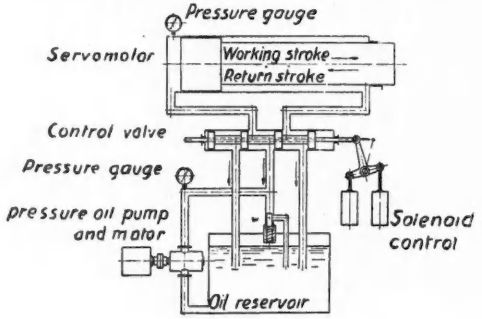


Fig. 1. Working circuit of power cylinder.

In steel works practice pressure-oil operation can with advantage be applied to various auxiliary equipment.

The simplest type of pressure oil mechanism in use is shown in Fig. 1. Here a pump draws oil from an oil reservoir and discharges it through an electromagnetically controlled valve into one or the other of the cylinder cavities of the power cylinder, according to the position of the control valve. As the illustration shows, the cylinder cavity at the right-hand side is greatly reduced by the employment of a piston rod of comparatively large diameter. This reduction in working cylinder volume as compared to that of the left-hand side cylinder cavity, causes the back stroke to take place at higher speed than the forward stroke. In addition to the safety valve, indicated in Fig. 1, a solenoid operated relief valve may also be employed, which opens a return line at the delivery side of the pump during idling.

A pressure-oil system of this type is installed in conjunction with an ingot-pusher, as shown in Fig. 2. Many such installations with thrusts ranging from 5 to 40 tons are in operation in Swedish steel works.

Fig. 3 (p. 106) shows a plant in which six ingot-pushers of the pressure-oil type are installed, with provision made for the installation of a seventh unit. The pressure-oil is delivered by two gear-type pumps, each pump serving three ingot pushers. This plant is of considerable dimensions, the distance between the pumps being some 85 meters. The larger pieces to be handled weigh 1,250 kg., and the individual furnace capacity is 43 tons. To move the entire charge of 43 tons, the ingot pusher must exert a thrust of 40 tons, which is considerably in excess of usual requirements.

The pressure-oil operated driving mechanism of a walking beam type furnace of 32 m. length is outlined in Fig. 4. The drive is equipped with two power cylinders which alternately go into action, thus producing a walking beam motion, the stroke of which may be adjusted between 300 and 500 mm. As will be seen

Fig. 2. **Ingot pusher of 40 t thrust and 1500 mm. strok e.**

Fig. 4. **Pressure-oil operated drive for walking beam furnace.**

Fig. 5.
Pressure-oil circuit of plant (Fig. 4).

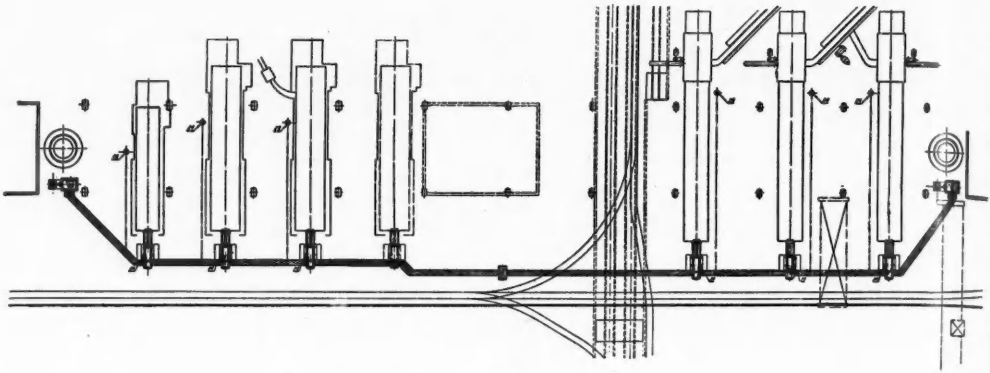


Fig. 3. **Plant with six ingot pushers.**

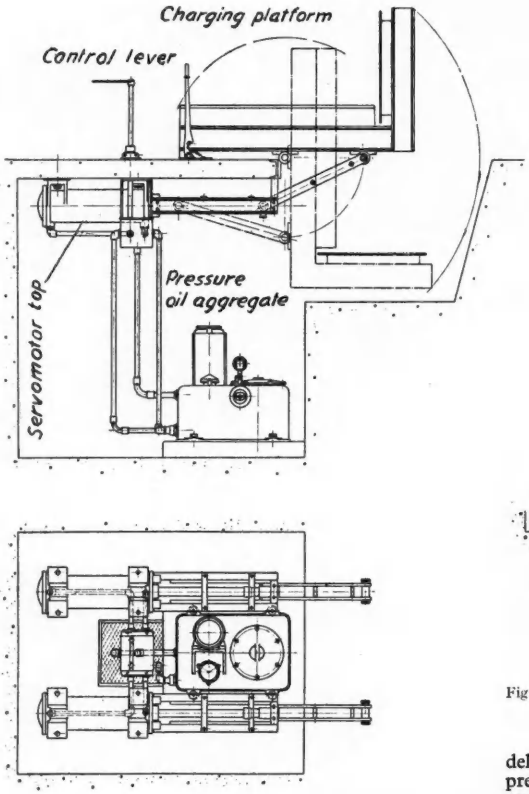


Fig. 7. **Charging device of wire-annealing furnace.**

from the illustration, the power cylinders, the control gear, and the oil reservoir form a single compact unit. The pressure-oil circuit of this drive, which completes a full operating cycle every 10 seconds, is shown diagrammatically in Fig. 5.

A three-high rolling mill stand equipped with pressure-oil operated roller tables is shown in Fig. 6. The manipulators (not shown) are likewise pressure-oil operated. The pressure-oil is furnished by four Imo pumps, two of which have a capacity of 35 litres per second at 20 atmospheres oil pressure. The speed of these pumps is 950 r.p.m. The other two pumps

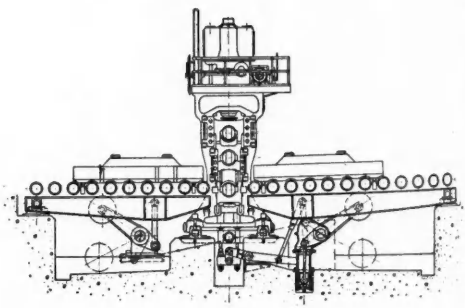


Fig. 6. **Pressure-oil operated tables of three-high rolling mill.**

deliver 5·5 litres of oil per second at 25 atmospheres pressure, the pump speed being 2850 r.p.m. All motions of the mechanism are controlled by a system of solenoid and motor operated valves. The mill itself is capable of handling 18 in. ingots of 2·5 tons weight.

Another application of the pressure-oil drive in a wire mill is shown in Fig. 7. This is a charging machine for the charging of coiled wire into an annealing furnace. After loading of the device, its platform is tilted by 90 deg. by means of two pressure-oil operated power cylinders.

METHODS FOR MEASURING THE TEMPERATURE OF LIQUID STEEL IN THE OPEN-HEARTH FURNACE.

By V. S. KOTSHO. (From *Stal*, No. 9/10, 1944, pp. 273-278, 8 illustrations.)

FOR the purpose of ascertaining the temperature of the liquid steel in the open-hearth furnace in some Russian steel works, samples of the steel are taken during the deoxidation stage by means of a ladle, and the temperature is then measured by means of an optical pyrometer or a thermocouple. Both methods have been investigated at the Dzerzhinsk and the Petrovsk Steel Works, and it has been established that with both of them a very large error was incurred, which amounted to 3·5 to 4·0 per cent, and sometimes to 5 per cent, of the actual temperature. Naturally this inaccuracy greatly diminishes the usefulness of these methods. It has been shown by another Russian investigator that in the measurement of the temperature of the molten steel in the ladle, it is necessary to introduce a correction of + 20 deg. C. to allow for the cooling of the metal in the ladle (often this error amounts to 30–45 deg. C.). The error due to the black body coefficient, which can only be estimated roughly, must also be added. The total error may thus be as large as 50–70 deg. C.

If a thermocouple is employed, each measurement of the metal temperature in the ladle takes about 10–15 seconds, and during this time the metal temperature drops considerably, so that the total error may be as much as 30–55 deg. C. Also, the magnitude of the error does not remain constant, as it is considerably influenced by: (1) the speed at which the metal flows from the furnace into the ladle; (2) the amount of slag in the ladle; (3) the amount of metal contained in the ladle; and (4) the weight and heat capacity of the ladle proper. Great interest therefore attaches to methods by which the temperature of the liquid steel in the open-hearth furnace itself can be measured.

COMBINED THERMOCOUPLE.

A special thermocouple for measuring the temperature of the bath was designed. This device consists of a jacket made of a piece of thick-walled steel tubing having a thickened end piece as shown in Fig. 1, this end piece to be immersed in the bath. The tip is preferably made of steel, but it can also be made of refractory material or graphite. In the head of the device a stopper is arranged through which a thin metal tube passes, which contains the thermocouple proper. The hot junction of the thermocouple, which projects some 90 mm., is protected by a thin quartz tube affixed to a holder of special design. This holder permits easy replacement of the quartz tube.

Prior to immersion of the hot junction of the thermocouple into the bath, the protecting tube is retracted in order to prevent the quartz tube from becoming covered with slag. The entry of slag or liquid steel into the tube is prevented by blowing compressed air through the tube. Upon immersion of the device in the liquid metal the hand-wheel (2) Fig. 1, is pushed forward until the quartz tube projects through the opening of the tip of the jacket some 60 mm., as indicated in Fig. 1a. There is no cooling action exerted by the compressed air stream discharged from the top of the tube, as the air stream penetrates only a few millimeters into the metal. This fact has been ascertained by test. The pressure of the compressed air should not be less than 0·5–1·0 kg./sq. cm. g. Tests have shown that the actual measurement with this device is easily and rapidly carried out.

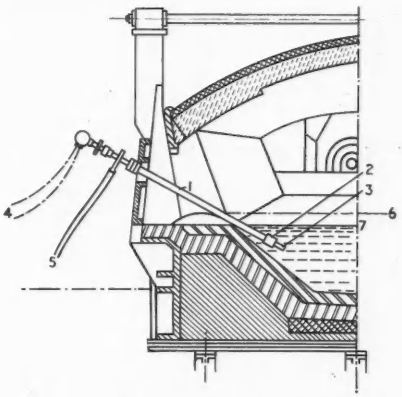


Fig. 2. **Thermocouple immersed in bath for purpose of measurement.**

(1) Thermocouple. (2) End piece. (3) Quartz tube. (4) Leads to galvanometer. (5) Compressed air hose. (6) Slag level. (7) Metal level.

Fig. 2 shows the device immersed in the bath of an open-hearth furnace. The thermocouple is connected with its measuring instrument by means of a flexible metal hose. This instrument has been used to measure the bath temperature in both basic and acid furnaces, and for carbon steel, as well as for alloy steel. The temperatures measured ranged from 1360 to 1670 deg. C. The following are typical test figures obtained on an acid electric furnace of 4·5 tons capacity :—

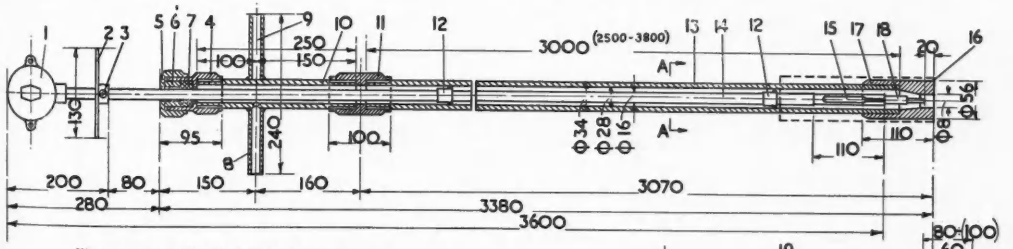


Fig. 1. **Combination thermocouple.**

(1) Terminal box. (2) Handwheel. (3) Adjusting screw. (4) Stuffing box. (5) Collar nut. (6) Nut. (7) Ring. (8) Nipple. (9) Handle. (10) Tube. (11) Coupling. (12) Coupling. (13) Tube. (14) Internal tube. (15) Holder. (16) End piece. (17) Screw thread. (18) Quartz tube. (19) Insulating tube. (20) Positive lead. (21) Negative lead. (22) Insulating capillary. (23) Hot junction.

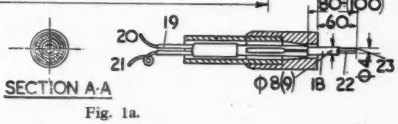


Fig. 1a.

(1) First 20 minutes after melting of the charge .. .. .. 1510 deg. C.
(2) Immediately before tapping .. 1630 „ „
(3) Temperature during tapping as measured by optical pyrometer 1535 „ „
Difference between (2) and (3) .. 95 „ „

The black body coefficient of the liquid steel was found to be $\epsilon = 0{\cdot}53$.

When measuring temperatures are in excess of 1620 deg. C., and where the metal is covered with a thin layer of slag, it is not easy to keep the end of the thermocouple from slagging, and the end of the tube may melt. In order to eliminate this danger, a greater length of the end piece, or better still, the entire end piece, should be made of graphite. Furthermore, the time required for carrying out the measurement should be shortened as much as possible. This can be achieved by reducing the thermal inertia of the hot junction of the thermocouple by means of a decrease in the diameter and the wall thickness of the protecting quartz tube. The relationship between the dimensions of the quartz tube and the thermal inertia was found as follows :—

Wall thickness of quartz tube, mm.	Time required in seconds for attainment of temperature equalization with different quartz tube diameters D(mm.).		
	D = 3	D = 4	D = 5
1·0–1·2	—	16–21	18–25
0·7–0·9	10–14	11–15	12–16
0·4–0·6	6–9	7–10	9–11

But in this respect it is of course impossible to exceed the limit given by consideration of mechanical strength of the quartz tube. The best results were obtained with a quartz tube of an external diameter of 4 mm. and a wall thickness of 0·7 mm., which requires a time of 10–11 seconds for temperature equalization.

The temperature rise characteristics of quartz tubes of 4 mm. diameter with different wall thicknesses are charted in Fig. 3. Here the shaded area indicates the permissable variation in the wall thickness, which lies within the range from 0·5 to 0·8 mm. thickness.

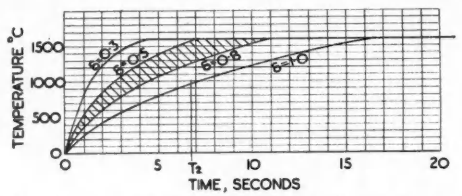


Fig. 3. **Temperature rise of quartz tubes of 4 mm. diameter with different thicknesses.**

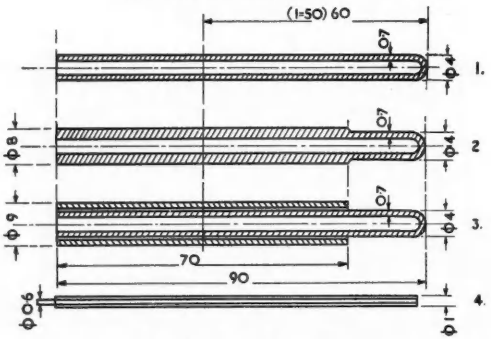


Fig. 4. **Different types of quartz tubes.**
(1) Thin walled tube. (2) Drawn-out tube. (3) Double tube. (4) Capillary. (*l*) Part of tube projecting from end-piece.

Various types of quartz tubes are shown in Fig. 4, where the distance "*l*" marks the depth to which the tube is immersed in the molten steel when carrying out the temperature measurement. The quartz tubes with the drawn-out ends 2 and 3 have proved to be sufficiently strong to allow reliable temperature measurement. The quartz capillary 4 serves as insulation for the wires of the thermocouple inside the quarz tubes 2 and 3.

It is recommended that a new protective tube should be used for each measuring operation. Any suitable type of indicating instrument may be used in conjunction with the thermocouple, the chief requirement being rapidity of indication. Highly accurate results will be obtained with the employment of a high-speed automatic potentiometer. Satisfactory results will also be achieved with galvonometers of the mirror type, but care must be taken to select a suitable type. Fig. 5 shows

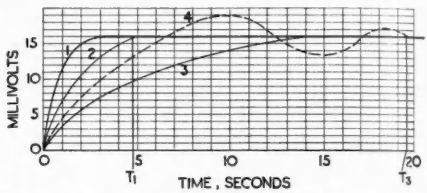


Fig. 5. **Movement of the needles of galvanometers of different type versus time.**
(1) Galvanometer with moving system of low inertia. (2) Ordinary type of galvanometer. (3) Galvanometer with suspended mobile system. (4) Galvanometer with damping.

the readings obtained with galvanometers of different types, where it is seen that galvanometers Nos. 1 and 2 are satisfactory, while Nos. 3 and 4 are unsuitable.

Two different types of thermocouples were used, namely, platinum-platinumrhodium and tungsten-molybdenum couples. The best results were obtained with the former type because of their large electromotoric force and their stable thermo-electric characteristics.

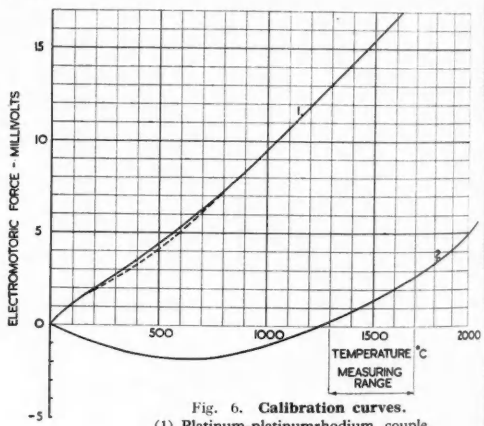


Fig. 6. **Calibration curves.**
(1) Platinum-platinumrhodium couple.
(2) Tungsten-molybdenum couple.

The electric characteristics of these two types of thermocouples are charted in Fig. 6, where the potential *vs* temperature curve of the tungsten-molybdenum couple is seen to be quite peculiar. At 1600 deg. C. the platinum-platinumrhodium couple is shown to possess an electromotoric force eight times that of the tungsten-molybdenum couple, but the rate of increase in electromotoric force of the platinum-platinumrhodium couple is only 1·7 times that of the tungsten-molybdenum type.

THEORETICAL INVESTIGATIONS INTO THE DISTORTION OF ELECTRIC IMPULSES IN VALVE CIRCUITS USED IN TELEVISION.

By J. HUBER. (From *Schweizer Archiv*, Vol. 11, Nos. 3 and 4, March and April, 1945, pp. 77-86, and 115-127, 30 illustrations).

I.—INTRODUCTION.

THE term " scanning " as used in this article means the process of dissecting a picture to be transmitted into points and the electrical measurement of the latter's brightnesses. The variable potential and current originating in the photocell during the scanning is called the picture signal. This picture signal is transmitted to the receiver over a " channel." In passing through an ideal channel the picture signal should not be distorted. In practice, however, distortion takes place and the degree to which it does occur is a fair measure of the quality of the television transmission. To determine the quality of a television transmitter, " test pictures " are usually transmitted which require a high performance of the transmission channel. A feature of these test pictures is the presence of the " black-white edge," i.e., sharp dividing lines between black and white.

The present article examines mathematically the valve circuits of a transmision channel as to their bearing on the distortions mentioned. In particular, the distortions of the picture signal of the black-white edge will be investigated. If an ideal scanning beam were used, the picture signal of a black-white edge would consist of a sharp potential or current surge. In practice, however, the surge requires a definite period of time, which, though short by itself, is nevertheless in the order of the time required for other processes in television. The valve circuits investigated consist mainly of a chain of electron valves and of four-terminal networks whose electrical properties vary with the applied frequency. The first task will be to define the elements of the circuit and to combine them in a way suitable for dealing with the problem set. The second task will be to establish functions which will put the electrical conditions on the input side in mathematical relation to those on the output side, and to discuss these functions.

The valve circuits in a television channel are mainly amplifiers and modulators. Fig. 1 shows an element of such a circuit consisting of an electron valve with a network in the anode circuit.

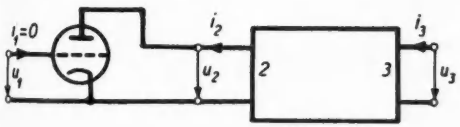


Fig. 1. **Basic circuit.**

The valve may have one or more grids. It has a more or less non-linear characteristic and a finite—if large—internal resistance. The properties of the four-terminal network in the anode circuit usually vary with the applied frequency.

The problem under discussion is to establish mathematical relations between u_2 and i_3 on the one side, with u_1 on the other.

II.—THE VALVE.

For the valve the following definition is valid :

$$i_2 = F(u_1 + D\,u_2) \quad .. \quad .. \quad (1)$$

and from this by definition :

$$\text{Slope } S = \left(\frac{\partial i_2}{\partial u_1}\right)_{u_2 = \text{constant}} = F'(u_1 + Du_2) \quad (2)$$

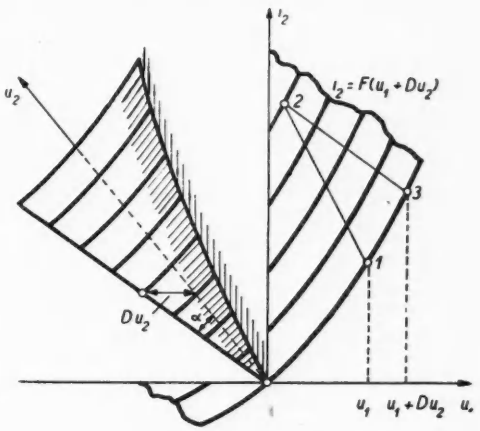


Fig. 2. **Valve characteristics.**

Internal conductivity

$$G_1 = \left(\frac{\partial i_2}{\partial u_2}\right)_{u_1 = \text{constant}} = DF'(u_1 + Du_2) \quad .. \quad (3)$$

Amplification factor

$$D = -\left(\frac{\partial u_1}{\partial u_2}\right)_{i_2 = \text{constant}} = \tan\alpha \quad .. \quad (4)$$

where F' is the derivative of F with respect to u_1.

With the known relation

$$SD\,\frac{1}{G_1} = 1 \quad .. \quad .. \quad .. \quad (5)$$

equations (2) and (3) can be simplified to

$$S = F'(u_1) \quad .. \quad .. \quad .. \quad (6)$$

$$G_1 = DF'(u_1) \quad .. \quad .. \quad .. \quad (7)$$

which, for valves with non-linear characteristics, are approximate equations, the error being the smaller the larger the conductivity of the anode circuit in relation to the internal conductivity G_1 of the valve.

III.—THE NETWORK.

(*a*) *The linear two-terminal network.*

A linear passive two-terminal network, by definition is, a network consisting of ohmic resistances, inductances and capacities, having only one pair of external terminals. The questions put in connection with a two-terminal network may be twofold.

(i) what are the properties of the current i_1 if a variable potential is applied to the terminals ;

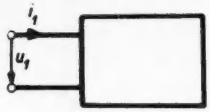


Fig. 3. **Two-terminal network.**

(ii) what are the properties of the potential u_1 if a variable current is made to flow through the network ?

In both cases, the function is a linear differential equation in which the known factor or its derivation appears as a disturbing function, for instance :

$$L\,(i_1\,(t)) = u_1\,(t). \quad .. \quad .. \quad (8)$$

If the disturbing function is a harmonic function of time ($\epsilon^{j\omega t}$) and if only permanent conditions are under investigation, the same equation may also be obtained in the conventional way by applying $j\omega L$ to each inductivity and $\dfrac{1}{j\omega C}$ to each capacity and by treating them like ohmic resistances.

If the disturbing function is an arbitrary function of time, equation (8) may be solved by using the Laplace-Transformation (L-Transformation)

$$f(p) = p\int_0^\infty \epsilon^{-pt}\,F(t)\,dt \quad .. \quad (9)$$

where p is a " subvariable " to the main variable t.

The solution to this equation shows that it is possible to calculate immediately the electrical conditions in a two-terminal network to which a potential or a current of whatever function of time is being applied, once the conditions following the application of one individual potential or current impulse are known.

(*b*) *The linear four-terminal network.*

A linear passive four-terminal network, by definition, is a network consisting of ohmic resistances, inductances and capacities having two pairs of external terminals. The behaviour of this network on application of arbitrary potentials or currents can be described by differential equations or, assuming that at $t = 0$ no energy was stored in the network, also by operational calculus.

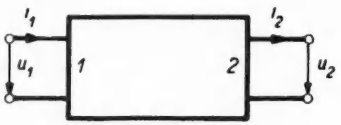


Fig. 4. **Four-terminal network.**

The problems to be solved in the case of a four-terminal network are as follows : Given are either the current or the potential at the input terminal. What are potential or current respectively at the input terminals or either potential or current at the output terminals ? The answers to these problems are again given by Laplace-transformation of the matrix form of the four-terminal network equations.

IV.—THE COMPLETE CIRCUIT (Fig. 1).

The problem of finding a mathematical relation between u_2 or i_2 on the one side and u_1 on the other is solved in 2 stages. First, u_2 or i_2 will be linked with u_1 ; subsequently, the conditions on the output terminals of the four-terminal network of the anode circuit will be calculated for the conditions already calculated on the input terminals.

With the equations developed for the two-terminal and four-terminal networks and the valve equation in the form :

$$u_2 = \frac{i_2 - F\,(u_1)}{DF'\,(u_1)} \quad .. \quad .. \quad (10)$$

the following equation is obtained :

$$i_2(t)\,(1 + Z_{22}\,(0)\,D\,F'\,(u_1)) =$$

$$- F\,(u_1) - DF'\,(u_1)\int_0^t Z'_{22}\,(t-\xi)\,i_2\,(\xi)\,d\xi \quad (11)$$

and, with $Z_{22}\,(0)\,DF'\,(u_1) = \delta(t)$: .. .. (12)

$$i_2(t) = \frac{F\,(u_1)}{1+\delta(t)} - \frac{\delta(t)}{1+\delta(t)}\int_0^t \frac{Z'_{22}\,(t-\xi)}{Z_{22}\,(0)}\,i_2\,(\xi)\,d\xi \quad (13)$$

which is actually a linear Volterra's equation of the type :

$$S(t) = H_1(t) + H_2(t)\int_0^t K\,(t-\xi)\,S\,(\xi)\,d\,\xi =$$

$$= H_1(t) + \int_0^t K\,(t-\xi)\,H_2(t)\,S(\xi)\,d\xi \quad .. \quad (14)$$

where $S\,(t)$ is the unknown and $H_1\,(t)$ the disturbing function. $K\,(t-\xi)\,H_2\,(t)$ is called the core of the equation ; it is asymmetric. The equation can be solved by progressive approximation. $Z_{22}(t)$ is called the " characteristical input impedance."

Similar equations can be developed for the potential $u_2(t)$.

Assuming that in the anode four-terminal network all inductances are taken out and all capacities short-circuited, $Z_{22}(t)$ becomes the " characteristical input impedance " corresponding to a frequency $\omega = \infty$; in this case equation (13) is reduced to

$$i_2\,(t) = \frac{F\,(u_1)}{1 + \delta(t)} = \frac{F(u_1)}{1 + Z_{22}\,(0)\,DF'(u_1)} \quad (15)$$

This equation for $i_2(t)$ can be called the dynamic characteristic of the circuit for ohmic resistance.

If it is assumed that the characteristic of the valve is linear for the range under investigation, the following equation is valid :

$$F(u) = S\,u_1 \quad .. \quad .. \quad .. \quad .. \quad (16)$$

and $$F'(u_1) = S = \text{constant slope} \quad .. \quad (17)$$

and equations (7) and (8) become exact. Equation (12) changes into :

$$\delta = Z_{22}\,(0)\,D\,S = Z_{22}\,(0)\,G_1 \quad .. \quad (18)$$

and (14) becomes :

$$S(t) = H(t) + \lambda\int_0^t K(t-\xi)\,S\,(\xi)\,d\xi \quad .. \quad (19)$$

which can be solved by Laplace-transformation, the solution being :

$$S(t) = H(t) + \lambda\int_0^t [K\,(t-\xi) + \lambda\,K^{(2)}\,(t-\xi) +$$

$$+ \lambda^2\,K^{(3)}\,(t-\xi) + \ldots]\,.\,H(\xi)\,d\xi \quad .. \quad (20)$$

V.—THE SCANNING DEVICE.

As already stated at the beginning of this article the transmission of a television picture begins with the photoelectric measurement of the brightnesses of individual elements of the picture by the scanning device.

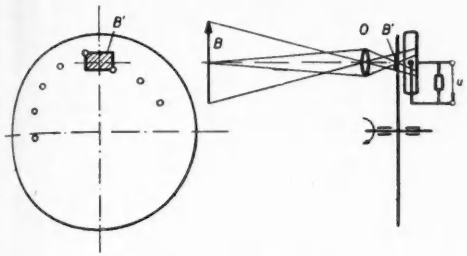


Fig. 5. **Scanning disc.**

The process of scanning can be carried out by various methods, the best known mechanical method using the scanning disc which is a rotating disc provided with a helical series of apertures (Fig. 5). The picture B to be transmitted is projected by a lense 0 into the plane of the disc (B'). As the disc moves round the apertures sweep across the area B' which represents the size of the picture to be scanned line by line and the luminous flux reaching the photocell at any instant is proportional to the brightness at the corresponding point of the picture. It should be noted that the voltage u at the photocell is a function of the luminous flux reaching the latter and thus a mean value for the area of one aperture of the disc. As the size of aperture obviously cannot be reduced below a certain limit, the picture signals are slightly distorted if compared with the picture itself. Details of the picture within the area of the scanning aperture cannot be reproduced exactly; the analyzing capacity of the scanning device is, therefore, strictly limited.

In the following, the analyzing capacity of a rectangular aperture will be investigated mathematically. For this purpose it will be assumed that the brightness H of the picture to be transmitted is constant in the direction perpendicular to the scanning line (Fig. 6).

$$H = H(x)\,;\ H(x<0) = 0\,. \qquad .. \qquad (21)$$

The picture itself is assumed to consist of elements of a width dx and of a height equal to the height of the aperture. Its brightness at a point x is $H(x)$. If the aperture moves uniformly in direction x, the abscissa x of a picture element can be interpreted as the time required by the front edge of the aperture to move from $x = 0$ to x. Thus the width T of the aperture may be

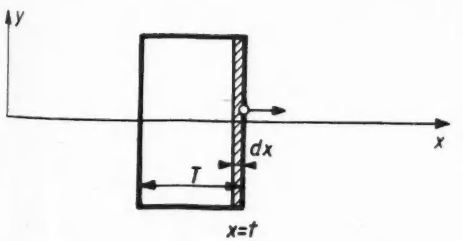


Fig. 6. **Rectangular scanning aperture.**

taken as the time a picture element stays behind the aperture. At the time t the total light flux reaching the photocell is proportional to the integral brightness of the picture elements which have been scanned by the front edge of the aperture in the interval between $t - T$ and t.

$$\phi(t) = \frac{C}{T}\int_{t-T}^{t} H(x)\,dx \qquad .. \qquad .. \qquad (22)$$

Assuming unity luminosity $H = 1$ over the whole picture, $\phi(t)$ becomes equal to C, which is therefore a constant factor indicating the properties of this optical system. Equation (22) is valid for $t \geq T$, while the corresponding equation for $t \leqq T$ is

$$\phi(t) = \frac{C}{T}\int_{0}^{T} H(x)\,dx \qquad .. \qquad .. \qquad (23)$$

Equations (22) and (23) can be transformed into

$$\phi(t) = C\int_{0}^{t} \frac{H(x) - H(x-T)}{T}\,dx \qquad .. \qquad (24)$$

which shows that the light flux transmitted through a rectangular aperture is the integral of the differential quotient of the luminosity distribution for the interval T. When T approaches 0, the differential quotient becomes the differential coefficient, and thereby

$$\phi(t) = C\,.\,H(t) \qquad .. \qquad .. \qquad (25)$$

Therefore, if the change of luminosity of B' in direction x is small enough to make the differential quotient over a width T practically equal the differential coefficient, the useful liminous flux is proportional to the brightness.

For investigating the conditions in the case of a "black-white edge," at $x = 0$ equations (22) and (23) are transformed by substituting $(t - \xi)$ for x into

$$\phi(t) = \frac{C}{T}\int_{0}^{t\leqq T} H(t-\xi)\,.\,d\xi \qquad .. \qquad (26)$$

$H(x)$ represents therefore a sudden leap of the luminosity by 1 (Unity leap). $H(x) = 1$ for $x>0$, $H(x) = 0$ for $x<0$, and therefore

$$H(t-\xi) = 1 \text{ for } \xi<t,\ H(t-\xi) = 0 \text{ for } \xi>t \qquad (27)$$

Equation (26) changes into

$$\left[\ \phi(t)\ \right]_{H(x)=\text{unity leap}} = E(t) = \frac{C}{T}\int_{0}^{t\leqq T} d\xi =$$

$$= C\,\frac{t}{T} \text{ for } t \leqq T\ , \text{ or}$$

$$= C \text{ for } t \geqq T \qquad .. \qquad (28)$$

$E(t)$ represents the useful luminous flux in the case of a black-white edge, and will in the following be called the "aperture transfiguration factor." It unequivocally describes the aperture, which means that the knowledge of it makes it possible at once to calculate $\phi(t)$ for any brightness distribution $H(x)$. Fig. 7 shows $E(t)$ for a rectangular aperture of unity transparency. The subfunction to $E(t)$ may be called $e(p)$ and can be calculated according to (9):

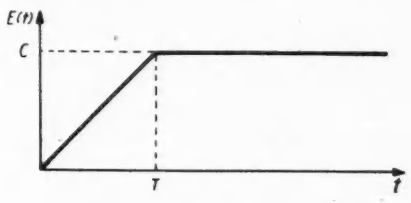


Fig. 7. **Transfiguration characteristic of a rectangular aperture of unity transparency.**

$$e(p) = \mathfrak{L}\,E(t) = p\int_0^\infty \epsilon^{-pt}\,E(t)\,dt$$

$$= Cp\int_0^T \epsilon^{-pt}\cdot\frac{t}{T}\,dt + Cp\int_T^\infty \epsilon^{-pt}\,dt$$

or, for $\mathfrak{Re}\ p>0$:

$$e(p) = C\frac{1-\epsilon^{-pT}}{pT} \quad .. \quad .. \quad (29)$$

Equation (29) represents, for $p = j\omega$, the complex aperture transfiguration factor. For a brightness distribution $\sin \omega x$ or $\cos \omega x$, the amplitude of the corresponding luminous flux, derived from this aperture configuration factor, is a function of the local frequency $\omega = \frac{2\pi}{x_0}$, namely,

$$e(j\omega) = C\frac{1-\epsilon^{j\omega T}}{j\omega T} = C\frac{\sin\frac{\omega T}{2}}{\frac{\omega T}{2}}\cdot\epsilon^{-j\omega\frac{T}{2}} \quad .. \quad (30)$$

$$\left|e(j\omega)\right| = C\left|\frac{\sin\frac{\omega T}{2}}{\frac{\omega T}{2}}\right| = C\frac{\left|\sin\pi\frac{T}{x_0}\right|}{\pi\frac{T}{x_0}} \quad (31)$$

Equation (31) shows the "frequency characteristic" of the aperture. x_0 is the cycle of the brightness distribution which was assumed to follow a sin or cos function. The "frequency characteristic" shows zero points for

$$x_0 = \frac{T}{n}, \text{ when } n = 1, 2, 3 \ldots \text{ (see Fig. 8)} \quad .. \quad (32)$$

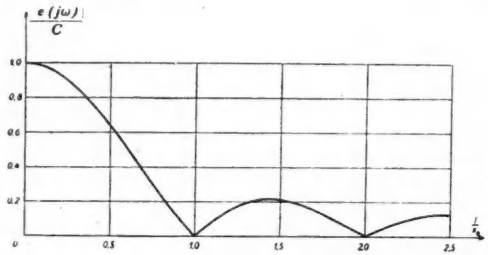


Fig. 8. **Frequency characteristic of a rectangular aperture of unity transparency.**

Assuming the existence of an ideal photocell (i.e. without electrostatic capacity), the equation describing the voltage of the picture signal of an ideal black-white edge is

$$u_s(t) = U\frac{t}{T} \text{ for } t \leqq T \text{ and } u_s(t) = U \text{ for } t \geqq T \quad (33)$$

Since the law of superposition can be applied to the calculation of linear networks, as are the two- and four-terminal networks described in paragraph III, it is useful to transform equations (33) so as to show u as a superposition of two increasing potentials (Fig. 9), namely:

$$u_s = U\frac{t}{T}\Big|_{t\geqq 0} - U\frac{t-T}{T}\Big|_{t\geqq T} \quad .. \quad .. \quad (34)$$

which is the equation for a picture signal representing a black-white edge and which forms the basis for the investigations into complex circuits.

For apertures of a different shape (circular, etc.), the transfiguration functions can be calculated in a similar way.

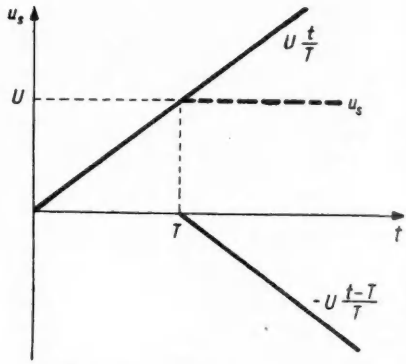


Fig. 9. **Picture signal of a black-white edge.**

VI.—R.C.-AMPLIFIER AND MODULATOR CIRCUITS.

Employing the formulae for the individual circuit elements arrived at in the preceding paragraphs, the author then analyzes mathematically the properties of the R.C.-amplifier circuit shown in Fig. 10 and of the modulator circuit shown in Fig. 11, and finally gives a detailed numerical example.

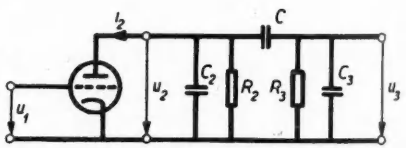


Fig. 10. **RC-amplifier circuit.**

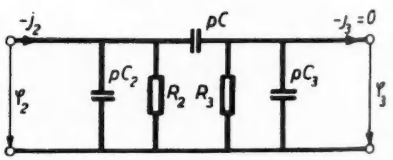


Fig. 11. **Modulator circuit.**

THE DISC SPRING.*

By K. WALTZ. (From *VDI-Zeitschrift*, Vol. 88, No. 47/48, Nov. 25th, 1944, pp. 643-646, 9 illustrations).

LATEST publications about Disc Springs which belong to the group of springs subjected to bending refer to their importance, qualities and applications only. In the following, the most important principles for manufacture and design are discussed.

Although the calculation of the disc spring in the simplest form is known since 1903, they found application only in few cases. For, the efficiency of disc springs, i.e. the proportion of output to input of work, was supposed to be low, and the disc spring needs a special guide, for instance, a tube or a bolt.

It has, however, been proved that disc springs show definite advantages even under alternating and repeated loads.

DESIGN OF THE DISC SPRINGS.

Figure 1 shows the various dimensions which decide the construction of disc springs. Various methods of

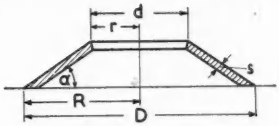


Fig. 1. Dimensions of a Disc Spring.

(Angle α exaggerated). Gauge of sheet s
Internal diameter d = 2r Angle α
External diameter D = 2R

calculating these springs have been developed, but generally the following two equations can be used to calculate the disc with sufficient accuracy, provided that the angle α does not exceed 7 deg., and that the stress on the inner edge (σ) is not in excess of 180 kg. per sq. mm.

$$\text{Load } P = \frac{s^2\,\sigma}{\varphi_2}\ \text{(kg.)} \quad .. \quad .. \quad (1)$$

$$\text{Elastic deflection } f = \varphi_1 \frac{R^2}{s}\,\sigma\ \text{(mm.)} \quad .. \quad .. \quad (2)$$

$$\text{Spring modulus } C = \frac{P}{f} = \frac{s^3}{\varphi_1\,\varphi_2 R^2}\ \text{(kg. per mm.)} \quad (3)$$

Figure 2 shows the values of the coefficients φ_1 and

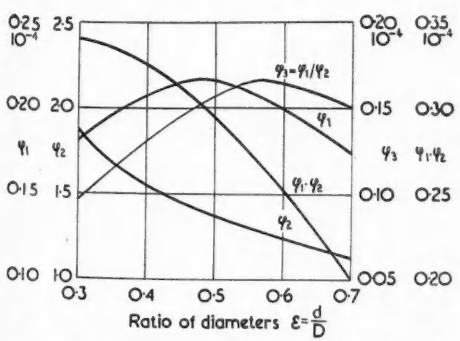


Fig. 2. **Coefficients for calculation of a disc spring.**

φ_2 plotted against the ratio $\epsilon = \frac{d}{D} = \frac{r}{R}$, as well as the product $\varphi_1\,\varphi_2$ and the ratio $\varphi_3 = \frac{\varphi_1}{\varphi_2}$. The maximum for φ_1 is at $\epsilon = 0{\cdot}475$.

If the ratio $\frac{s}{D}$ decreases below 0·03, there may be the danger that the inner edge starts cracking under the influence of radial stresses. If on the other hand the ratio $\frac{s}{D}$ exceeds 0·06 the three above mentioned equations cannot be applied as the conditions of stress in the disc no longer correspond to the suppositions from which they result.

From equations (1) and (2) follows :

$$\text{Input of work } A = \frac{Pf}{2} = \frac{sR^2\,\sigma^2}{2}\ \text{(kg. mm.)} \quad .. \quad (4)$$

whereby $\varphi_3 = \frac{\varphi_1}{\varphi_2}$. The optimum will be obtained at $\epsilon = 0{\cdot}575$ (see Fig. 2).

Generally the input of work of a single unit is expressed by the equation

$$A = k\,\frac{V\sigma^2}{E} \quad .. \quad .. \quad .. \quad (5)$$

where V (mm³) is the volume and E (kg. per sq. mm.) the modulus of elasticity. The optimum for k will be 0·5 if stress σ is equally distributed over the entire section under stress.

From equations (4) and (5) can be deducted :

$$\varphi_3\,\frac{sR^2}{2} = k\,\frac{V}{E} \quad .. \quad .. \quad .. \quad (6)$$

As the volume of a disc is expressed by

$$V = (R^2 - r^2)\,\pi s \quad .. \quad .. \quad (7)$$

it follows from (6) and (7) :

$$k = \frac{E}{2\pi\,(1 - \epsilon^2)}\,\varphi_3\,; \quad .. \quad .. \quad (8)$$

consequently the degree of utilization of the material depends on the ratio of the diameters ϵ, which, according to Fig. 2, shows the optimum of work input at $\epsilon = 0{\cdot}575$. In this case $k = 0{\cdot}08$ is less favourable than for example in the case of a spiral spring with circular section ($k = 0{\cdot}25$) or for a leaf spring of rectangular cross section and parabolic thickness ($k = 0{\cdot}167$). It follows that the frequently advantageous application of disc springs is not based on their economic use of the material.

Disc springs are noteworthy for their high fatigue resistance as a consequence of the supporting effect resulting from the bending stress. In addition disc springs can be applied to strong forces in cases where other forms of springs could not be used because of their excessive size. Besides, several disc springs in series have a high damping effect, which could not be obtained with helical springs without additional elements.

Figure 3 shows the use of two and more discs combined to one group. Figure 4 is the reproduction of the force-deflection diagram of various groups of disc springs ($D = 36$ mm. $d = 16{\cdot}4$ mm. $s = 2{\cdot}45$ mm. and $\alpha = 4$ deg. 30′). Calculation as well as experiments show

* These springs are also known under other terms, e.g. Cup springs, Belleville springs.—Ed., E.D.

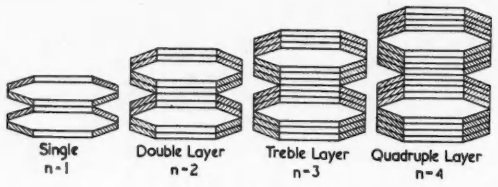


Fig. 3. **Disc springs, single and in several layers.**

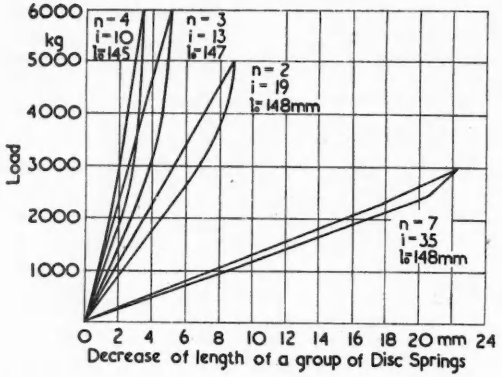


Fig. 4. **Force-deflectio diagram of disc springs in different layers.**

l_0 = normal length of a group of disc springs
n = number of disc springs combined to one group.
i = number of groups.
Load
Decrease of length of a group of disc springs.

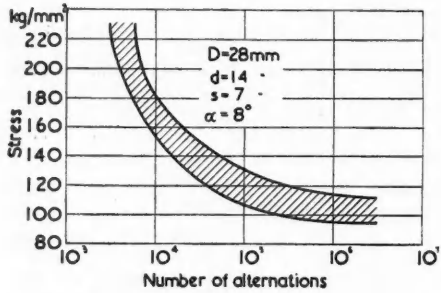


Fig. 5. **Scattering of the Wöhler curves of a single disc spring made of material III in table I, under alternating loads.**

that a group of n discs, put into each other absorbs n times the force of a single disc spring for producing the same deflection. Consequently the spring modulus C as well as the input of work increase to the n fold value for a group of n discs. The deflection, however, of the single discs in a group is the same throughout and therefore the fatigue does not increase in spite of the n times heavier load.

Equation (4) proves that the same effect can be obtained by increasing the width s of the material n times. In this case however, the spring modulus C will be increased n^3 times (see equation (3).

Supposing the same extent of deflection, the single disc inside a group under excessive load is less stressed than a disc of a spring consisting of one disc only. In addition it must be taken into account, that one of the discs in a group may break. If in this case a shell or a tube prevents the parts from falling off, the spring continues to work with its length unchanged. Only its damping effect might be increased slightly.

The efficiency decreases with the number of discs within a group, but no clear results can be derived in practice from this theoretical rule, as the efficiency is subjected to fluctuations which cannot be anticipated with sufficient accuracy, e.g. those caused by friction on guide pins.

The maximum stress depends on the kind of load, the method of production, and the material and its heat treatment ; it can rise up to a fatigue resistance of 300 kg. per sq. mm. on the outer edge. A combination of static and dynamic loading would lower its maximum the more the higher the share of the alternating stress. Repeated stress, i.e. stress originating from fluctuations between zero and a maximum, must be considered the most unfavourable case of loading. Figure 5 shows the range of scattering of the lifetime curves in case of repeated stress for disc springs made of material III in Table I.

TABLE I. MATERIALS FOR DISC SPRINGS.

Material	Composition C %	Si %	Mn %	P %	S %
I	0·5 bis 0·7	<0·45	0·5 bis 0·8	<0·035	<0·035
II	0·55 bis 0·70	1·6 bis 1·9	0·6 bis 0·8	<0·035	<0·035
III	0·55 bis 0·65	0·9 bis 1·1	1·0 bis 1·2	<0·030	<0·030

MATERIAL AND FINISH.

Material I (see Table I) should be used generally. It shows good results with regards to fatigue and alternating stresses. Material II shows good qualities under very high fatigue stresses, but should be avoided for alternating stresses because of the decarbonization of the edges frequently resulting from the high contents of silicon. Material III is specially suited for high alternating stresses.

Springs should be made from hot rolled sheets or cold rolled strips. The latter is more liable to fracture in the case of springs under alternating stress by the strongly marked direction of the fibre, whereas variations in the thickness of sheets should be equalized by grinding the sheet on the internal side of the disc. In extreme cases of alternating stress and small safety factor, the sheets should be rolled twice in two directions vertical to each other, and should be wet ground on both sides.

Discs for alternating loads should be stamped oversize and then ground or turned to finished size. The disc hole should be punched and reamed. This procedure is not necessary for discs under purely static stress.

The discs should be hardened out of a salt solution in oil or better, in a warm solution of 180 deg. C. By choosing the correct tempering temperature a Rockwell C hardness of 45 to 50 should be obtained, exceeding these figures only in the case of fatigue stress. Ball blasting is used for disc springs under alternating stress only. They must be protected afterwards against corrosion. Electro-zinc plating has proved a success whereas chromium plating has resulted in shorter life time owing to the segregation of hydrogen during the galvanic process. The chromium layer, as consequence of its brittleness, shows a tendency of peeling off.

Before use the discs should be overloaded as the remaining deformations result in favourable internal stresses.

A case hardened or heat treated bolt should be used as a guide for a group of disc springs. Variations of the dimensions, resulting from deviations of the thickness of the material or of the height of the discs, should be equalized by the use of special shims.

CLASSIFIED ABSTRACTS

Important articles which, for lack of space, cannot be given in full in our main editorial columns, are published as shorter abstracts in this section. Subscribers may obtain photostat copies of all original articles at cost.

AIRCRAFT

Aircraft Performance Testing and Reduction by the " iw " Method.

By J. J. GREEN, M.E.I.C., Chief Research Aeronautical Engineer, Air Transport Board, Ottawa, Ont., and F. S. NOWLAN, United Air Lines, Chicago, Ill., U.S.A. (From *The Engineering Journal*, Vol. 28, No. 12, Dec., 1945, pp. 758-771, 783, 14 illustrations.)

THE so-called " iw " method for the determination of aircraft performance in a standard atmosphere from light tests carried out from day to day under non-standard conditions has become very popular in recent years. The basis of the method is by no means new, having been indicated as a performance reduction system as long ago as 1920 (" Applied Aerodynamics " by L. Bairstow).

Consider an aircraft in level flight in a non-standard atmosphere. Let W be the instantaneous weight of the aircraft, V the steady level true air speed for a brake horsepower output P, and N the rotational speed of the propeller. Assume that the altitude is such that the air density is p. The " iw " method consists in determining, from this flight test data, what the new speed and necessary power output would be if the aircraft were flying at the *same angle of attack* but at sea level in the standard atmosphere and at some new weight, at which performance figures are required.

If level flight speeds are measured in the tests at various altitudes and over as wide a range of engine powers as possible, then a family of curves can be drawn for sea level in a standard atmosphere showing the relationship between level speed and engine power (at the standard weight) for various values of N_{1w} ' the sea level propeller speed. For this to be accomplished, of course, the power P developed by the engine in the flight tests must be known. The use of a torquemeter gives the most reliable method for determining the power, but reliable engine power charts have proved very satisfactory.

From the level speed versus power curves at sea level, the level speed at altitude in a standard atmosphere can be determined by the reverse process, for any required power setting. Using power curves to give the engine power P at height, P_{1w} can be calculated for the appropriate height. The P_{1w}, V_{1w} curve then gives the corresponding value of V_{1w} and hence V the true speed.

The treatment of climb performance is done in a similar fashion.

ELECTRIC GENERATORS AND MOTORS

A Carbon-Pile Speed Governor.

By C. T. BUTTON, Assoc. A.I.E.E. (From *Electrical Engineering*, Vol. 65, No. 1, Jan., 1946, Trans. pp. 4-7, 6 illustrations.)

THE governor referred to was designed for application on aircraft inverters to hold the speed and, consequently, the output frequency substantially constant. Such a governor may be used on a variety of d-c motors.

The problem of governing the speed of a d-c motor is not unlike that of controlling the voltage of an a-c generator—both involve sensitive variation of excitation of a d-c field in response to variation of the controlled quantity. For example, a two per cent change in either voltage or frequency of the generator output should produce perhaps a 50 per cent change in excitation of generator or motor field, respectively. Such a device may be regarded broadly as a rate-of-change amplifier with a high amplification ratio around the control point.

A contact-making regulator is the simplest form of such a device and has been used widely for both voltage and speed regulation. The limitations of a contact-making regulator are well-known ; and the advantages of a carbon pile for frequency regulation are exactly the same as for voltage control. Briefly, the functional advantage of the carbon pile is that it varies field excitation by infinitesimal steps so as to achieve a point of stability on a smooth curve for any given conditions. The ordinary carbon-pile voltage regulator when viewed as an amplifier may be compared with a triode amplifying tube by drawing an analogy between carbon-pile resistance and cathode-to-plate impedance ; and between solenoid voltage and grid voltage.

The carbon-pile voltage regulator easily may be used to control speed by introducing either an inductance or a tuned circuit in series combination with the solenoid of the carbon pile so that it will be sensitive to frequency change. While there are many ways of producing varying pressure on a carbon pile in response to change in frequency or speed, probably the most direct and efficient way is to utilize thrust resulting from centrifugal force on rotating weights supported by springs.

ELECTRONICS

Induction Electron Accelerator.

(From *General Electric Review*, Vol. 48, No. 11, November, 1945, pp. 49-50, 2 illustrations.)

FOR the production of Roentgen rays corresponding to voltages in excess of a few million, the induction electron accelerator is today the most promising device. It has proved successful in the 20-million and the 100-million-volt-sizes and can presumably be used for still higher voltages.

The tube is a hot-cathode high-vacuum device consisting of a hollow toroid of glass or other insulating material about 18 inches in diameter for the 20-million-, and 6 feet in diameter for the 100-million-volt sizes. The electrons from a hot filament are electrostatically focused and are accelerated with some 20,000 volts or more in a direction tangential to the axis of the toroid. By means of a time-varying magnetic field, they are further accelerated and focused and constrained to follow a circular path within the tube. In the 100-million-volt machine they encircle the field 250,000 times in 1/240 second, receiving on an average a 400-volt push each time around. After travelling in this way in a circular path for about 800 miles and receiving the energy which they would have had if they had passed between two electrodes having a potential difference of 100 million volts, they are caused to leave their circular orbit and impinge upon a tungsten target where they produce Roentgen rays. Or they may be caused to leave the circular orbit at any desired earlier time in the magnetic cycle, thus producing Roentgen rays corresponding to any voltage up to 100,000,000.

The electron current in the beam is very low compared with that which has ordinarily been used in Roentgen tubes. It is of the order of a microampere, and

THE LITTLE TUBE THAT DID A BIG WAR JOB

This little tube is estimated to have saved more than 2,000,000 gallons of petrol! A simple job—an air vent tube inserted into a 'Jerrican'—but it halved emptying time and eliminated loss of fuel by spilling. We have made over 40,000,000 of these air vent tubes—only one of our many war jobs.

It is just such a simple device—produced at our plants at high speed—which may help enormously in your future production programme. We can supply Tru-Wel electrically welded steel tubes to your specification in any quantity. Our mass production methods are controlled to ensure absolute uniformity in the first and the millionth length—no worries on the assembly benches. Why not investigate further the possibilities of Tru-Wel tubes?

Tru-Wel electrically welded steel tubes are mass-produced to your requirements: identical in lightness and strength; in concentricity and evenness of wall-strength; identical in composition so that all can be manipulated with equal ease and at full-power speed.

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depends to a great extent on the frequency employed for the time-varying magnetic field.

The tubes may be built either as unitary structures or in sections cemented together and continuously pumped. A 100-million-volt consists of 16 pie-shaped sections of heat-treated pyrex glass having elliptical cross-section 8 in. by $4\frac{7}{8}$ in. The ends of the sections are ground flat and smooth and to the correct angle, and the joints, coated on the outside with Glyptal paint, are vacuum tight.

The induction electron accelerator, as a multi-million-volt Roentgen-ray source, will be extensively used in scientific research and should also find therapeutic and industrial applications.

PLASTICS

Friction Welding of Thermoplastics.

By K. ROSE, Engineering Editor, Materials & Methods. (From *Materials & Methods*, Vol. 22, No. 6, Dec., 1945, pp. 1728-1730, 5 illustrations.)

A NEW joining method promises to become a valuable addition to present assembly and fabricating methods. It has been called " friction welding " because it consists of heating the material by friction to local fusion, then quickly pressing together the surfaces to be welded. It is applicable only to thermoplastics, and at present has been found to give satisfactory results only in joining plastics of the same composition, although no serious theoretical obstacle stands in the way of use of the method for joining dissimilar thermoplastics. It has also been used on a laboratory scale for joining plastics to wood and to metal.

An important advantage of friction welding is its extreme simplicity. No special machinery is required. The method consists of bringing the two plastic surfaces together, with one or both moving rapidly, and with just sufficient pressure applied to cause enough friction to create appreciable heat ; then, as the surfaces reach the fusing point, the two pieces are pressed firmly together and the power causing motion cut off. The films of molten plastics solidify at once, and the two pieces are fused together.

In welding, the prepared piece of plastic is fastened in the chuck of a high-speed drillpress, speed lathe, vibrator, or similar machine. Moving at high speed—about 6000 r.p.m. for small work—it is brought against the piece to which it is to be welded. The second piece can be held in a vice on the drillpress table, or fixed correspondingly in whatever other machine is used. The two pieces of work are then brought into light contact. Within a few seconds fumes rise from the material, and the operator will observe a " flowing " of the resin as the surfaces in contact reach the fusion point. The switch operating the machine is then turned off, and the pieces are immediately pressed firmly together, using the spindle feed lever on the drillpress or the handwheel on the lathe. Since only the surfaces have been heated appreciably in the few seconds elapsed, the film of liquid material solidifies at once. A thin, brittle flash is squeezed out by the consolidating pressure, and can be broken away without difficulty. The entire process will have required less than 30 seconds.

Kriston—A New Thermosetting Resin.

(From *Plastics*, Vol. 3, No. 6, December, 1945, pp. 40, 42, 43, 115, 116, 9 illustrations)

A NEW liquid thermosetting resin which can be cast in inexpensive moulds without pressure or high heat and with a minimum shrinkage, the new allyl ester type resin, called ' *Kriston* ', is said to possess an outstanding combination of electrical, mechanical, and optical properties. Thus, it is expected to find optical, electrical, chemical, and decorative uses both alone and with other materials in laminate form.

Because Kriston can be ground and polished much more easily than glass and because it is very clear and has a higher index of refraction than most optical glass, it is a promising material for lenses, prisms, reflectors, and refractors. Its electrical properties are expected to lead to numerous uses. Preliminary tests indicate that the polymer is relatively stable electrically over the temperature range of—10 deg. C to 140 deg. C. It appears to rank below polystyrene and polyethylene at room temperature.

Resistance to fats, oils, and greases suggest many uses in the food and petroleum industry. Resistance to corrosion makes the new material applicable for use as or in valves, pumps, agitators, controls, etc. In general, the stability of Kriston when exposed to a wide variety of reagents offers the basis for extended applications in chemical processing and manufacturing.

Allyl esters have been known for years, but wide commercial utilization as plastics materials has been handicapped by the volume change during polymerization. This deficiency has been overcome in Kriston, which has a shrinkage on cure of only 3·25—4 per cent. The allyl ester (monomer) is a pure chemical compound, crystal clear and of syrupy consistency. It has a high molecular weight with corresponding low vapour pressure and high boiling point. The specific gravity varies from 1·26 at 10 deg. C. to 1·178 at 130 deg. C. The viscosity changes from 50,000 centipoises at 22 deg. C. to 12 at 126 deg. C. The refractive index is 1·5483 at 28 deg. C. It is soluble in ketones, esters, ethers, alcohols and aromatic hydrocarbons.

Laboratory experience has indicated that a standardized procedure can be established for preparation of the monomer-catalyst system. Benzoyl peroxide is pulverized (with care) and is then passed through a 100 mesh screen. The required quantity of the screened catalyst, usually 1 per cent. by weight, is suspended in the Kriston monomer as evenly as possible by stirring. The mass is then heated to approximately 70 deg. C., preferably by use of an oven or jacketed vessel. The catalyzed monomer is then ready to use.

If an ordinary casting is to be made, the catalyzed monomer is poured into the mould and the whole heated in an oven or otherwise to 70 deg. C., until all air bubbles have risen to the surface and broken. Heating is then continued at 70 deg. C. until polymerization has been completed. The casting is then ready for removal from the mould.

RADIO ENGINEERING

A New High-Efficiency Linear Amplifier.

By S. T. FISHER, M.E.I.C., F. T. Fisher's Sons Ltd., Consulting Engineers, Montreal. (From *The Engineering Journal*, Vol. 29, No. 1, Jan., 1946, pp. 13-18, 10 illustrations.)

THE problem of devising a high-efficiency linear amplifier for a carrier wave modulated in amplitude by a speech wave is one of the most important in all radio engineering because of the large use of radio transmitters for aircraft and other mobile uses. The following proposal is believed to be a basically correct approach to the problem.

The anode dissipation in a vacuum tube goes to a low value when either the anode current is reduced to a low value without exceeding the allowable anode voltage, or the anode voltage is reduced to a low value without exceeding the maximum anode current. The wave form which fulfils both these conditions will be transmitted with maximum efficiency, and this wave form is a square-topped wave. If signalling systems were called upon to transmit only such wave forms, of constant amplitude,

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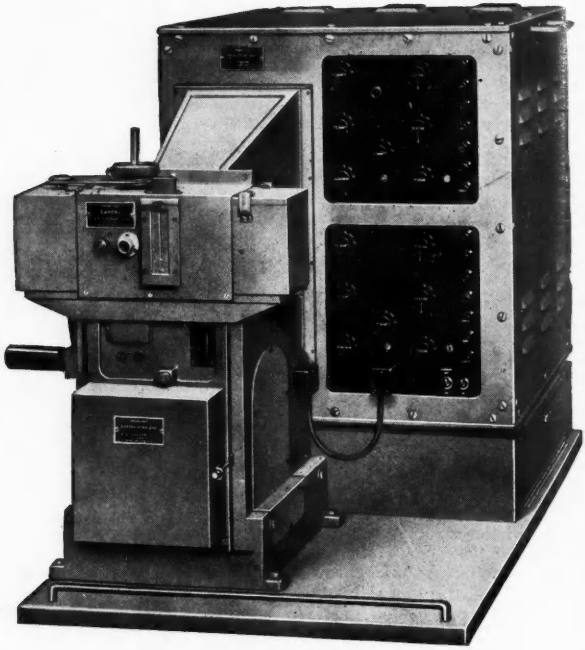
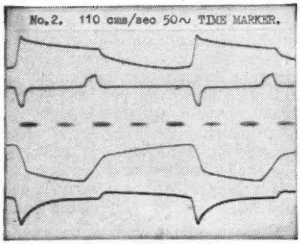


This equipment is the latest development in the use of sealed-off Cathode Ray Tubes in Multiple Recording technique as pioneered by Cossor. The robust design is intended to make it suitable for field tests, including Traction Recording, Marine and Motor testing and other industrial investigations. The equipment comprises a Model 440 Twin Tube Unit using two 4½" dia. Double Beam Tubes, a Model 423 Film Camera Unit and a Model 430 Drive Unit. It provides simultaneously four independent records plus one central electro-optical time marking on 70 mm. unperforated film or paper. By means of interchangeable gears and/or motors a film speed range of 0.005-250 cms/sec. is obtainable with a tube spot writing speed up to 2 kms/sec. Two or three units may be mounted one above the other and driven by the single Drive Unit to provide up to a maximum of twelve simultaneous records plus three time/event markings.



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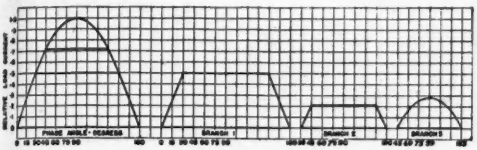
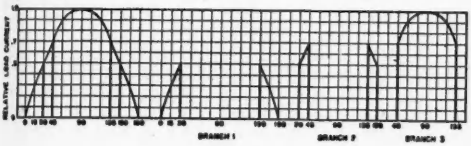
we would then have linear power amplifiers which would operate with an efficiency of the order of 90 per cent, using conventional tubes.

The illustration shows two ways in which this can be done. The wave can be divided into a number of sections, horizontally (a) or vertically (b). The method



(a)



(b)

proposed consists of the following steps : (1) Subdivision of the wave into components that approximate the optimum, rectangular wave form ; (2) Amplification of these components separately in vacuum-tube amplifiers ; (3) Recombination of the separate components in the output circuit of the amplifiers so as to reproduce the original wave at a higher power. Thus high-efficiency amplification is achieved by dividing the wave on an amplitude basis into several sections, in practical cases, about three ; transmitting each of the sections through a power amplifier whose peak allowable current is that of the section being transmitted, and then combining the three amplifier branches to a common load circuit so that the original wave form is again obtained.

ROLLING MILL PRACTICE

Die Rolling.

By H. F. MARQUARDT, Superintendent of Rolling, Republic Steel Corp., Buffalo, N.Y. (From *Iron and Steel Engineer*, Vol. 23, No 1, Jan., 1946, pp. 68-71, 5 illustrations.)

IN 1930 Republic Steel, under the Thomas H. Sloan patents, began the manufacture of flashless die roll blanks. These blanks, while not as close to a finished article as the flash blanks, leave sufficient metal for the forger to make the largest part of a projected article and at the same time greatly reduce the metal where the forger requires less.

The die roll engineer is furnished with a description of the part to be made as well as a knowledge of the machinery, and methods to be used in making the part. Then he lays out the design for a blank of proper length, weight and shape. Stock is distributed so that there is sufficient to make the large sections of the forging but is reduced for the smaller dimensions. Proper allowances must be made for extrusion or contrusion, the degree of transition from dimension to dimension, and for the amount of reduction to the leader bar which may vary from little to over 75 per cent.

The design is then given to the die roll shop. Here two sleeve rolls are placed on one mandrel, and the periphery of the rolls is scribed to mark the points where variations in the grooves are to be located. In scribing the two rolls at one time, the same keyway is used, and later little adjustment is needed to get rolls in time or registration in the mill. The rolls now go to the lathes where a lead groove is cut for each pass. This is a groove corresponding to the smallest dimension desired, and is cut by lathe because it is quick and economical. The larger parts of the groove are cut on vertical or horizontal milling machines, or perhaps on a fly cutting machine, and require considerable time, skill, and patience. Templates are furnished by the layout man, so that proper transitions from section to section are maintained. In the final operation the rolls go to a grinding stand where the grooves are ground and polished to micrometer dimensions.

By allowing for a gate between blanks it is possible to redress most rolls several times. This is done by reducing the diameter of the rolls enough to remove worn parts of the grooves.

SCIENTIFIC INSTRUMENTS AND PRECISION TOOLS

Design of Supersensitive Balancing Equipment.

By K. F. FRANK, Sperry Gyroscope Co. Inc. (From *Machine Design*, Vol. 18, No. 2, February, 1946, pp. 109-114, 180, 7 illustrations.)

THERE is a distinction between ordinary requirements and precision balancing of small high-speed rotors operating at speeds from 3,000 to 100,000 r.p.m. where in some instances unbalance moments of 0·000,002 ounce-inch must be detected. Hand balancing of high-speed rotors is a slow and tedious operation requiring considerable skill. It is also difficult or practically impossible when balancing rotors that have bearings with high noise level, because the sense of feel cannot discriminate against bearing vibration. Heretofore the required degree of balance has been difficult to obtain due to the lack of suitable testing equipment.

In the balancing equipment discussed, three units are employed, namely : the balancing unit on which the necessary electrical impulses for determining magnitude and position of unbalance are generated ; the amplifiers in which these signals are mixed, filtered, amplified and indicated on a meter ; and a drilling unit, where the unbalances indicated can be removed accurately.

The balancing unit is composed of a shock-mounted bedplate on which two headstocks, a photoelectric cell and exciter-lamp assembly, and an air nozzle valve block are mounted. These are adjustable to accommodate various size rotors.

When an unbalanced rotor is mounted in the cradle holder and rotated at balancing speed, the armature, which is attached to the cradle holder and lies on the same axis as that of the rotor to be tested, will oscillate about the centre pole of the vibration pick-up. The amount of magnetic flux travelling up and down the centre pole depends upon the position of the ring armature. The radius of oscillation about the centre pole is proportional to the rotor unbalance, and the voltage generated in the pick-up coil is proportional to the radius of oscillation. Extraneous vibrations which are due to bearing imperfections are of different frequencies than the rotor speed and are filtered out in the amplifier by an electrical filter. This type of pick-up does not restrain the unbalance motion, since its direct coupling to the unbalance vibration is magnetic, yet it does not introduce lost motion. This feature is essential in microbalancing and does not interfere with the low resonant suspension of the rotor under test.

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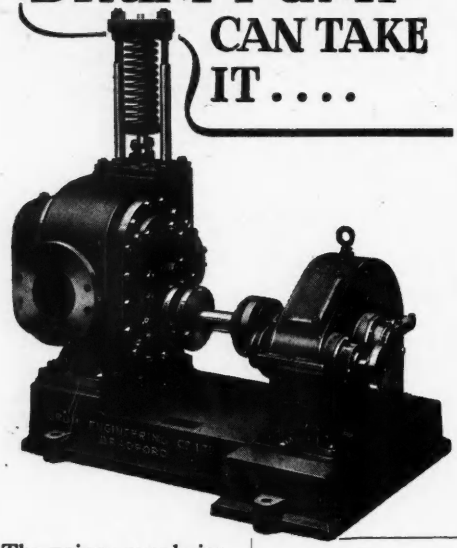

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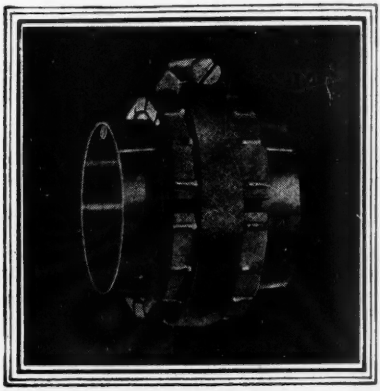

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